

DESIGN OF A TEST INSTALLATION FOR  
INVESTIGATION OF REYNOLDS' NUMBER  
EFFECT ON GAS TURBINE PERFORMANCE

---

WILLOUGHBY MERCER  
LEROY P. SMITH

Library  
U. S. Naval Postgraduate School  
Monterey, California







(  
Mont 162  
8854



REYNOLDS' NUMBER EFFECT ON GAS TURBINE PERFORMANCE,  
DESIGN OF A TEST INSTALLATION FOR INVESTIGATION OF

by

Omdr. Willoughby Mercer, USN  
B.S. in Eng'g., Harvard Univ., 1936

and

Lt. Cmdr. Leroy P. Smith, USN

Submitted in Partial Fulfillment of the  
Requirements for the Degree of

Master of Science

at the

Massachusetts Institute of Technology  
1949

RESEARCH REPORT NO. 1000  
OF THE NATIONAL BUREAU OF STANDARDS

Thesis  
M52

19

U.S. GOVERNMENT PRINTING OFFICE  
WASHINGTON, D.C. 20540

1954

U.S. GOVERNMENT PRINTING OFFICE

U.S. GOVERNMENT PRINTING OFFICE  
WASHINGTON, D.C. 20540

U.S. GOVERNMENT PRINTING OFFICE

1954

U.S. GOVERNMENT PRINTING OFFICE  
WASHINGTON, D.C. 20540

Massachusetts Institute of Technology  
Cambridge, Massachusetts  
May 17, 1949

Professor Joseph S. Newell  
Secretary of the Faculty  
Massachusetts Institute of Technology  
Cambridge, Massachusetts

Dear Professor Newell:

We take pleasure in submitting herewith a thesis entitled "Reynolds' Number Effect on Gas Turbine Performance, Design of a Test Installation for Investigation of", in partial fulfillment of the requirements for the degree of Master of Science in Aeronautical Engineering.

Massachusetts Institute of Technology  
Cambridge, Massachusetts  
Aug 27, 1949

Professor James E. Smith  
Department of the Biology  
Massachusetts Institute of Technology  
Cambridge, Massachusetts

Dear Professor Smith:

The book "Genetics in relation to the development of the individual" is written in a style which is both interesting and instructive. It is a book which should be read by all biologists. The book is written in a style which is both interesting and instructive. It is a book which should be read by all biologists. The book is written in a style which is both interesting and instructive. It is a book which should be read by all biologists.



## ACKNOWLEDGMENTS

The authors have been assisted considerably in the preparation of this thesis by the cooperation of various persons on the staff of the Gas Turbine Laboratory, the Mechanical Engineering Department and the Aeronautical Engineering Department. Professor E. S. Taylor has contributed unstintingly with his advice and experience, without which the authors would have been sorely handicapped. The requested advice of and analysis by Professor A. R. Rogowski relative to the feasibility of applying an Eddy Current Brake as a dynamometer was greatly appreciated. Mr. F. Lustwerk of the Gas Turbine Laboratory provided valuable information relative to his experience in use and operation of the wind tunnel. Mr. A. H. Redding of the Westinghouse Electric Corporation contributed an afternoon of his time, and provided a basis for the final choice of dynamometer type. He was instrumental in providing drawings and specifications of a similar device owned by the company. To such individuals, mentioned or otherwise, the authors wish to extend their deeply-felt gratitude.

1. Acknowledgment of Power Measurement by a Dynamometer

2. Power Measurement System

3. Acknowledgment of Design Department of the Gas Turbine

4. Power Measurement System and Power Measurement

5. Power Measurement System

Department of Pathology  
University of Toronto  
Sept 17, 1929

Professor George E. Smith  
Department of the Faculty  
University of Toronto  
Toronto, Ontario

Dear Professor Smith:

The new literature in relation to the 'Hepatic' and 'Biliary' system is of great interest to me. I have been particularly interested in the work of the American and English writers on the subject of the 'Hepatic' and 'Biliary' system. I have been particularly interested in the work of the American and English writers on the subject of the 'Hepatic' and 'Biliary' system.

Sincerely,  
W. L. H. H.

W. L. H. H.  
W. L. H. H.

W. L. H. H.  
W. L. H. H.

## ACKNOWLEDGEMENTS

The authors have been assisted considerably in the preparation of this thesis by the cooperation of various persons on the staff of the Gas Turbine Laboratory, the Mechanical Engineering Department and the Aeronautical Engineering Department. Professor E. S. Taylor has contributed unstintingly with his advice and experience, without which the authors would have been sorely handicapped. The requested advice of and analysis by Professor A. R. Rogowski relative to the feasibility of applying an Eddy Current Brake as a dynamometer was greatly appreciated. Mr. F. Lustwerk of the Gas Turbine Laboratory provided valuable information relative to his experience in use and operation of the wind tunnel. Mr. A. H. Redding of the Westinghouse Electric Corporation contributed an afternoon of his time, and provided a basis for the final choice of dynamometer type. He was instrumental in providing drawings and specifications of a similar device owned by the company. To such individuals, mentioned or otherwise, the authors wish to extend their deeply-felt gratitude.





## TABLE OF CONTENTS

<u>Section</u>		<u>Page</u>
1.	Introduction	1
2.	Summary	2
3.	Discussion	4
4.	Instrumentation	16
5.	Assembly of Equipment	20
6.	Operating Instructions and Precautions	24
 <u>Appendices</u>		
A	Dynamics of the Turbine Shaft Extension	27
B	Determination of Turbine Test Operating Conditions	31
C	Determination of State of Air at Turbine Inlet	33
D	Determination of Reynolds' Numbers	35
E	Determination of Turbine Power Output	36
F	Analysis of the Characteristics of a Flat Disc Running in Water	37
G	Analysis of the Characteristics of a Cylinder Partially submerged in Water	40
H	Determination of Power Absorption in a Centrifugal Water Pump	43
I	Thrust Bearing Design	45
J	Determination of Torque Capacity of the Plain Tapered Joint between Coupling and Shaft Extension	48
K	Air Safety Valve Spring Design	50
 <u>References</u>		 53

# TABLE NO. 1

Page	Section	Page
1	Introduction	1
2	Summary	2
3	Materials	3
4	Method	4
5	Results	5
6	Discussion	6
7	Conclusions	7
8	References	8
9	Appendix	9
10	Index	10
11	Table of Contents	11
12	Figure 1	12
13	Figure 2	13
14	Figure 3	14
15	Figure 4	15
16	Figure 5	16
17	Figure 6	17
18	Figure 7	18
19	Figure 8	19
20	Figure 9	20
21	Figure 10	21
22	Figure 11	22
23	Figure 12	23
24	Figure 13	24
25	Figure 14	25
26	Figure 15	26
27	Figure 16	27
28	Figure 17	28
29	Figure 18	29
30	Figure 19	30
31	Figure 20	31
32	Figure 21	32
33	Figure 22	33
34	Figure 23	34
35	Figure 24	35
36	Figure 25	36
37	Figure 26	37
38	Figure 27	38
39	Figure 28	39
40	Figure 29	40
41	Figure 30	41
42	Figure 31	42
43	Figure 32	43
44	Figure 33	44
45	Figure 34	45
46	Figure 35	46
47	Figure 36	47
48	Figure 37	48
49	Figure 38	49
50	Figure 39	50
51	Figure 40	51
52	Figure 41	52
53	Figure 42	53
54	Figure 43	54
55	Figure 44	55
56	Figure 45	56
57	Figure 46	57
58	Figure 47	58
59	Figure 48	59
60	Figure 49	60
61	Figure 50	61
62	Figure 51	62
63	Figure 52	63
64	Figure 53	64
65	Figure 54	65
66	Figure 55	66
67	Figure 56	67
68	Figure 57	68
69	Figure 58	69
70	Figure 59	70
71	Figure 60	71
72	Figure 61	72
73	Figure 62	73
74	Figure 63	74
75	Figure 64	75
76	Figure 65	76
77	Figure 66	77
78	Figure 67	78
79	Figure 68	79
80	Figure 69	80
81	Figure 70	81
82	Figure 71	82
83	Figure 72	83
84	Figure 73	84
85	Figure 74	85
86	Figure 75	86
87	Figure 76	87
88	Figure 77	88
89	Figure 78	89
90	Figure 79	90
91	Figure 80	91
92	Figure 81	92
93	Figure 82	93
94	Figure 83	94
95	Figure 84	95
96	Figure 85	96
97	Figure 86	97
98	Figure 87	98
99	Figure 88	99
100	Figure 89	100
101	Figure 90	101
102	Figure 91	102
103	Figure 92	103
104	Figure 93	104
105	Figure 94	105
106	Figure 95	106
107	Figure 96	107
108	Figure 97	108
109	Figure 98	109
110	Figure 99	110
111	Figure 100	111
112	Figure 101	112
113	Figure 102	113
114	Figure 103	114
115	Figure 104	115
116	Figure 105	116
117	Figure 106	117
118	Figure 107	118
119	Figure 108	119
120	Figure 109	120
121	Figure 110	121
122	Figure 111	122
123	Figure 112	123
124	Figure 113	124
125	Figure 114	125
126	Figure 115	126
127	Figure 116	127
128	Figure 117	128
129	Figure 118	129
130	Figure 119	130
131	Figure 120	131
132	Figure 121	132
133	Figure 122	133
134	Figure 123	134
135	Figure 124	135
136	Figure 125	136
137	Figure 126	137
138	Figure 127	138
139	Figure 128	139
140	Figure 129	140
141	Figure 130	141
142	Figure 131	142
143	Figure 132	143
144	Figure 133	144
145	Figure 134	145
146	Figure 135	146
147	Figure 136	147
148	Figure 137	148
149	Figure 138	149
150	Figure 139	150
151	Figure 140	151
152	Figure 141	152
153	Figure 142	153
154	Figure 143	154
155	Figure 144	155
156	Figure 145	156
157	Figure 146	157
158	Figure 147	158
159	Figure 148	159
160	Figure 149	160
161	Figure 150	161
162	Figure 151	162
163	Figure 152	163
164	Figure 153	164
165	Figure 154	165
166	Figure 155	166
167	Figure 156	167
168	Figure 157	168
169	Figure 158	169
170	Figure 159	170
171	Figure 160	171
172	Figure 161	172
173	Figure 162	173
174	Figure 163	174
175	Figure 164	175
176	Figure 165	176
177	Figure 166	177
178	Figure 167	178
179	Figure 168	179
180	Figure 169	180
181	Figure 170	181
182	Figure 171	182
183	Figure 172	183
184	Figure 173	184
185	Figure 174	185
186	Figure 175	186
187	Figure 176	187
188	Figure 177	188
189	Figure 178	189
190	Figure 179	190
191	Figure 180	191
192	Figure 181	192
193	Figure 182	193
194	Figure 183	194
195	Figure 184	195
196	Figure 185	196
197	Figure 186	197
198	Figure 187	198
199	Figure 188	199
200	Figure 189	200
201	Figure 190	201
202	Figure 191	202
203	Figure 192	203
204	Figure 193	204
205	Figure 194	205
206	Figure 195	206
207	Figure 196	207
208	Figure 197	208
209	Figure 198	209
210	Figure 199	210
211	Figure 200	211
212	Figure 201	212
213	Figure 202	213
214	Figure 203	214
215	Figure 204	215
216	Figure 205	216
217	Figure 206	217
218	Figure 207	218
219	Figure 208	219
220	Figure 209	220
221	Figure 210	221
222	Figure 211	222
223	Figure 212	223
224	Figure 213	224
225	Figure 214	225
226	Figure 215	226
227	Figure 216	227
228	Figure 217	228
229	Figure 218	229
230	Figure 219	230
231	Figure 220	231
232	Figure 221	232
233	Figure 222	233
234	Figure 223	234
235	Figure 224	235
236	Figure 225	236
237	Figure 226	237
238	Figure 227	238
239	Figure 228	239
240	Figure 229	240
241	Figure 230	241
242	Figure 231	242
243	Figure 232	243
244	Figure 233	244
245	Figure 234	245
246	Figure 235	246
247	Figure 236	247
248	Figure 237	248
249	Figure 238	249
250	Figure 239	250
251	Figure 240	251
252	Figure 241	252
253	Figure 242	253
254	Figure 243	254
255	Figure 244	255
256	Figure 245	256
257	Figure 246	257
258	Figure 247	258
259	Figure 248	259
260	Figure 249	260
261	Figure 250	261
262	Figure 251	262
263	Figure 252	263
264	Figure 253	264
265	Figure 254	265
266	Figure 255	266
267	Figure 256	267
268	Figure 257	268
269	Figure 258	269
270	Figure 259	270
271	Figure 260	271
272	Figure 261	272
273	Figure 262	273
274	Figure 263	274
275	Figure 264	275
276	Figure 265	276
277	Figure 266	277
278	Figure 267	278
279	Figure 268	279
280	Figure 269	280
281	Figure 270	281
282	Figure 271	282
283	Figure 272	283
284	Figure 273	284
285	Figure 274	285
286	Figure 275	286
287	Figure 276	287
288	Figure 277	288
289	Figure 278	289
290	Figure 279	290
291	Figure 280	291
292	Figure 281	292
293	Figure 282	293
294	Figure 283	294
295	Figure 284	295
296	Figure 285	296
297	Figure 286	297
298	Figure 287	298
299	Figure 288	299
300	Figure 289	300
301	Figure 290	301
302	Figure 291	302
303	Figure 292	303
304	Figure 293	304
305	Figure 294	305
306	Figure 295	306
307	Figure 296	307
308	Figure 297	308
309	Figure 298	309
310	Figure 299	310
311	Figure 300	311
312	Figure 301	312
313	Figure 302	313
314	Figure 303	314
315	Figure 304	315
316	Figure 305	316
317	Figure 306	317
318	Figure 307	318
319	Figure 308	319
320	Figure 309	320
321	Figure 310	321
322	Figure 311	322
323	Figure 312	323
324	Figure 313	324
325	Figure 314	325
326	Figure 315	326
327	Figure 316	327
328	Figure 317	328
329	Figure 318	329
330	Figure 319	330
331	Figure 320	331
332	Figure 321	332
333	Figure 322	333
334	Figure 323	334
335	Figure 324	335
336	Figure 325	336
337	Figure 326	337
338	Figure 327	338
339	Figure 328	339
340	Figure 329	340
341	Figure 330	341
342	Figure 331	342
343	Figure 332	343
344	Figure 333	344
345	Figure 334	345
346	Figure 335	346
347	Figure 336	347
348	Figure 337	348
349	Figure 338	349
350	Figure 339	350
351	Figure 340	351
352	Figure 341	352
353	Figure 342	353
354	Figure 343	354
355	Figure 344	355
356	Figure 345	356
357	Figure 346	357
358	Figure 347	358
359	Figure 348	359
360	Figure 349	360
361	Figure 350	361
362	Figure 351	362
363	Figure 352	363
364	Figure 353	364
365	Figure 354	365
366	Figure 355	366
367	Figure 356	367
368	Figure 357	368
369	Figure 358	369
370	Figure 359	370
371	Figure 360	371
372	Figure 361	372
373	Figure 362	373
374	Figure 363	374
375	Figure 364	375
376	Figure 365	376
377	Figure 366	377
378	Figure 367	378
379	Figure 368	379
380	Figure 369	380
381	Figure 370	381
382	Figure 371	382
383	Figure 372	383
384	Figure 373	384
385	Figure 374	385
386	Figure 375	386
387	Figure 376	387
388	Figure 377	388
389	Figure 378	389
390	Figure 379	390
391	Figure 380	391
392	Figure 381	392
393	Figure 382	393
394	Figure 383	394
395	Figure 384	395



## 1.0 INTRODUCTION

1.1 It is of vital concern to aviation interests to be able to make predictions of gas turbine performance with due regard to the effects of geometric scale and/or changes in altitude, which are directly related to Reynolds' Number. Due to the large quantities of air involved in the gas turbine cycle in actual practice, it is extremely expensive to construct wind tunnels, or other such test facilities, for the calibration or testing of such power units at varying Reynolds' Number. As a stop-gap measure, it has been the practice of designers to estimate performance of proposed engines on the basis of geometric and dynamic similarity to smaller or similar existing engines. Altitude performance has been predicted from sea level tests on actual engines in terms of dynamic similarity in speeds, temperature and pressure levels, etc. These predictions generally have disregarded the possibility that component efficiencies may vary for the conditions of such extrapolations. Cascade tests have provided a two-dimensional basis for correcting for induced aerodynamic losses, with fairly well substantiated empirical correction factors to three-dimensional conditions, (ref. (a)). However, airfoil profile losses, which are more directly related to Reynolds' Number, have not been so accurately determinable, especially over the possible range of operation at altitude.

1.2 The Gas Turbine Laboratory at MIT has in its possession a small scale turbo-jet engine manufactured by Westinghouse Electric Corp., namely the Model XJ32-WB-4 (X9.5B). This engine provides a single-stage turbine with a blade-tip diam. of 8.1 in. and a root diam. of 5.1 in. The variable-density wind tunnel at the above activity provides a test facility in which the performance of such a turbine could be studied in detail, over





a range of Reynolds' Numbers from 253,500 to 25,350 based on blade chord and turbine inlet conditions. (see Appendix D). This would extend the range of investigation reported in ref. (a) by approximately 25 per cent on the upper limit and 33 per cent on the lower limit of that of ref. (a). It is the purpose of this project to provide a design for the required test installation and associated equipment so that the above facilities may be used in the investigation of the effect of Reynolds' Number variation on turbine performance.

## 2.0 SUMMARY

2.1 Figures 3 to 26 show the general details of the resulting design. The necessary assembly and detail drawings for existing Westinghouse parts are available in the Gas Turbine Library and are described in ref. (h). It will be seen that provisions have been made for a horizontal installation in the low speed branch of the variable density wind tunnel on the second floor of the Gas Turbine Laboratory, MIT. The turbine wheel has been located as close as possible to an elbow adapter in the tunnel so that the power take-off could be taken through the elbow. This results in a condition whereby the air stream is delivered to the turbine very shortly after the turn through the elbow, but any other installation which would provide an external location for the dynamometer would necessitate excessively long extension shafts, which would be unacceptable from dynamic considerations. However, due to the rather severe acceleration and diffusion in the airstream through the compressor diffuser which was used to facilitate mounting, it is believed that flow at entrance to the turbine nozzles will be sufficiently uniform, in both





direction and degree of turbulence, as to be representative of actual "hot" running conditions at the turbine.

2.2 The installation shown will permit safe operation of the turbine at speeds from zero to somewhat in excess of 18200 rpm. (see Appendix A for shaft extension dynamics). The latter speed corresponds by dynamic similarity to an actual ("hot") running speed of 34000 rpm. (see Appendix B). Maximum desired running speed is 18,220 rpm. Static inlet pressure as delivered by the tunnel is controllable from 22.4 psia to 2.24 psia with mass flow of 7.005 to .7005 lbs./sec., permitting power variation in the ratio 10 to 1 at any turbine speed. Total temperature at tunnel compressor outlet is controllable from 80° to 100° F., corresponding to a static temperature range of from 73° to 93° F. at the turbine inlet. The possible range of investigation will be from a Reynolds' Number of 25,350 to 253,500.

2.3 Assembly directions for the test installation will be found in Sections 5.0 to 5.26 inclusive. Detailed operating instructions have been based on the various considerations of the proposed design, and on safety. They will be found in Sections 6.0 to 6.34 inclusive. It is to be noted especially that operation at speeds over 18200 rpm. will most probably be safe, since that represents the critical speed for the new turbine shaft extension, assuming zero fixity at the ends. The assumption is considered to be very conservative. All appendices have been prepared for a top speed of 20,000 rpm., at which it is recommended that the safety governor mentioned in Section 3.5 be set, thereby providing a margin of 1,780 rpm. over maximum desired operating speed, for adjustment purposes during running.





### 3.0 DISCUSSION

3.1 In providing a facility for measurement of performance of the turbine of the WM X9.5B Turbo-Jet engine, the first and most obvious need was for a device capable of absorbing and measuring an output of 250 hp. at 20,000 rpm. (see Appendix F), with possible variation in output in the ratio of 10/1 at any given speed. Such devices are not presently available at MIT, nor are they procurable commercially except, probably, on a developmental basis. Therefore it became necessary to design such a dynamometer locally. Several basic techniques were considered, all of which would lend themselves to direct coupling with the turbine (a very desirable and necessary feature), and are presented in the following paragraphs:

3.11 A plain disc running submerged in water, to absorb the required power through skin friction and circulation effects. The variation in power absorption could be controlled by "water level" (amount of water) on the disc, since any water in the disc chamber will be thrown as far out on the disc as possible by centrifugal effects. This technique was discarded when it was shown (Appendix F) that, although the maximum power could easily be absorbed by one submerged disc of small O.D., there would very likely be instability in operation at low power input at high speeds. The location of the inner radius of the water was very critical under such conditions. (Power varies approximately with fifth power of radius, third power of speed).

3.12 The above led to consideration of the possibility of using a simple cylinder rotating about a vertical axis in water of variable

1.1. The purpose of this study is to investigate the effects of the proposed system on the performance of the system.

1.2. The proposed system is designed to improve the performance of the system by reducing the time taken to process the data.

1.3. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.4. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.5. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.6. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.7. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.8. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.9. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.10. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.11. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.12. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.13. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.14. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.15. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.16. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.17. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.18. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.19. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.20. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.21. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.22. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.23. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.24. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

1.25. The system is designed to be able to handle large amounts of data and to be able to process the data in a timely manner.

depth. Power absorption would vary linearly with the submerged length of the cylinder (Appendix G). However, Appendix G shows that a shaft of excessive length would be required if any degree of stability were to be achieved at low power settings. Its vibration characteristics would probably be unacceptable, as seen from the analysis for the turbine shaft extension (Appendix A). Furthermore, a pressure-tight enclosing cylinder would be required to prevent the water from being thrown away uncontrollably in excess of that amount required for cooling. This would probably result in a violent circulation in the container, with possible instability, not readily analysed mathematically. For these reasons, the technique was considered to be not usable, and was discarded.

3.13 Power absorption by pumping water through a centrifugal water wheel. Since the water leaving the wheel would be thrown away or used for recirculation, input would probably be measured most easily by some sort of torque-meter on the connecting shaft. Water would be introduced into the wheel axially as near the axis as possible, and its momentum would be changed by producing tangential velocity at the rim. Appendix H shows that for reasonable flows, tangential velocities would be such as to cause unacceptable erosive action on any enclosing casing with which it might come in contact. A visit to the Gas Turbine Division of Westinghouse at their South Philadelphia Works revealed that that company had conducted tests to determine the severity of such erosion. They directed a jet of water normal to a 1/16-in. thick stainless steel plate mounted on a wheel



[illegible]

1. The first of these is the fact that the Commission has not yet received any information from the Government of the United States regarding the activities of the Committee for the Liberation of the Americas (CLA) in the United States. The Commission is therefore unable to determine whether the CLA is engaged in any activities which might be considered as a threat to the security of the United States.

which produced a relative velocity of 1000 ft./sec. The plate was severed within a period of five seconds. Although it is considered possible that other materials could be found which would show better durability in this desired application, it would be optimistic to believe that one could be found which would have acceptable service life. Therefore this technique was discarded.

3.14 A device which has shown very satisfactory results locally as a power absorption device is the Eddy Current Brake. This device consists of a metal disc rotating in electrically induced magnetic fields. The heat produced in the plate by the eddy current generation is carried away by water jets, usually directed through the cores of the electromagnetic coils. Ref. (b) gives the operating characteristics for such brakes with wheels of magnetic materials. More desirable high-speed characteristics are obtained when non-magnetic materials are used in the wheels. Although there has been no local experience with such brakes at the desired top operating speed, Prof. A. R. Rogowski of the Mechanical Engineering provided an extrapolation of some of his test results on brakes of several steels at speeds up to 10,000 rpm. This extrapolation predicted the following design provisions for the subject problem:

Material	Disc Size (Diam.)	No. Coil Pairs	Amp-turns/coil pair
18-8 Stainless Steel	18"	4	14,000
18-8 Stainless Steel	14"	5	14,000
Steel (Boiler Plate)	14"	8.4	14,000

However, it is shown in Appendix F that the necessary flow of water required for normal cooling in a water brake was 12.71 gal./min.. In the





Eddy Current Brake it is believed that the transmission of heat to the cooling water would be considerably less effective than in the simple water brake, resulting in a much higher flow requirement. This water would be subjected to considerable centrifugal pumping on the brake wheel surface, at radii of not less than 7-in., making for considerable unmeasurable losses. Further, windage losses alone on a wheel of such size would not be acceptable if not measurable. These problems could probably be obviated if power input were to be determined by torque measurements in the shaft, but such measurements are of questionable accuracy. Further, there would be the above-mentioned erosion problem of collecting water moving at high velocities. This technique was shelved therefore, in favor of one which showed more promise, discussed below.

3.15 During the above-mentioned visit to Westinghouse, it was learned that that organization has been using successfully a Froude-type water brake for applications similar to ours. This device is in effect a 100%-slip hydraulic clutch, with power absorption controllable by varying the amount of water in the casing, cooling to be provided by continuous replacement. Fig. 1 provides typical performance characteristics obtained by Westinghouse. The plot was obtained by extrapolation of the test data for much larger but slower similar brakes, adjusting scale by the fifth-power rule for power variation with radius, and the third-power rule for variation with rotational speed. Actual test results agreed very satisfactorily with the predictions on this basis. The characteristics desired in this project could be obtained by similar





extrapolation of the wheel and stator dimensions by the same rules. The company has delivered to M.I.T. the detailed drawings and performance data for this brake. They pointed out that in no way did they guarantee the design, performance, or safety of the device, nor are they on the market to sell such equipment. It is considered that it is within the capabilities of local facilities to manufacture such a device, using the general details provided by Westinghouse. It could be adapted in such a way that a "family" of characteristics similar to that of Fig. 1 could be obtained simply by providing different sets of power wheels and stators, with adapters which would give the proper casing voids. Although the production cost of such a brake is considerably greater than it would have been for some of the techniques described above, this is considered to be justified by the characteristics obtained and their reliability, and the stability and controllability of the device over the desired range.

3.2 In the interests of economy it was decided to use as much of the available turbo-jet engine as possible in the desired installation, to minimize the number of new parts which would have to be made. The construction of the engine permitted a complete separation of components just behind the compressor at the shaft coupling and forward face of the compressor diffuser flange. This left the compressor diffuser, combustion chamber, bearings support assembly, and turbine inlet nozzles available as a complete assembly in which the turbine sub-assembly could be installed complete with two radial roller bearings. The lubrication distribution system for the two bearings is integral with the resulting unit. However, since the thrust bearing for the entire turbo-jet unit





was incorporated in the compressor assembly, it became necessary to provide a replacement when the forward part of the engine was discarded. This has been provided external to the tunnel, and has been incorporated on the coupling of the turbine shaft extension, as shown on Fig. 3. in order to minimize turbine shaft extension length. (See Appendix I for analysis of Thrust Bearing requirements.) A sealed bearing is used to eliminate the lubrication problem and the possibility of contaminating the tunnel through the tunnel seal. The fuel nozzles and combustion chamber liner were removed and replaced with new inner fairings (Figs. 15 & 16) to conduct the airstream to the turbine with as little loss and turbulence as possible.

3.21 Normal lubrication of the two roller bearings (Nos. 2 and 3 of the turbo-jet engine) is by an oil-air mist produced by a pump (Westinghouse Part No. 23F530-2) and Mixer (Westinghouse Part No. 14G210-line 1) working off the accessory section. Since the accessory section has been discarded along with the forward part of the engine, it will be necessary to mount the pump and mixer somewhere external with an auxiliary drive for the pump (an electric motor at 1750 rpm. would be adequate). This should be mounted as close to the test installation as possible. Air for the mixer can be drawn from the room through a reducing valve by which pressure in the bearing support will be controllable. The scavenging system described below will provide the pressure differential required for flow. Ref. (h) shows the necessary piping and connections, which should be duplicated as nearly as possible in the test installation. The oil-air mist line should be introduced through the turbine support flange in the tunnel by means of a fixed gas-tight fitting.





3.22 In normal operation the oil-air mist is discharged from the bearings into the exhaust and thrown away. However, it is not permissible to discharge any contamination into the wind tunnel system because of cleanliness requirements when Schlieren apparatus or Interferometers are to be used. Therefore, some sort of an auxiliary scavenging system had to be devised. It was seen that the turbine bearings and assembly could be isolated within the compressor diffuser and turbine bearing support assembly. A labyrinth seal for the turbine shaft and a gasket is mounted on the after end of the bearing housing in place of the old radiation shields (Westinghouse Part Nos. 14G963-1&2), which were discarded. The forward end is sealed by means of a new nose cone, (Fig. 21), which seals on the diffuser casting in place of the former diffuser cone and seal (Westinghouse Part No. 19E269 line 1&2), and on the new shaft extension by a labyrinth seal incorporated in the new nose cone. Sharp crests are provided in the seals to provide for "running in" to minimum clearance on the shaft. The rubber gasket at the after end is needed to cover existing screw holes and other cutaways in the after bearings housing flange, and to effect the seal between the bearings housing and the bearings support. It is to be made oversize relative to the I.D. of the support tube, so that the necessary deformation during assembly will effect the seal. An inward flow must be established at all seals and joints, to ensure that no oil gets into the tunnel. Scavenging lines cut into the bearings support (see Fig. 17) will satisfy this requirement when connected to a large-capacity ejector available in the laboratory. The combustion chamber must be turned from its normal installed position relative to the diffuser so

1. The first step in the process of developing a new product is to identify a market need. This is done by conducting market research, which involves gathering information about the target market's needs, preferences, and buying habits. This research can be done through a variety of methods, including surveys, focus groups, and interviews with potential customers. Once the market need has been identified, the next step is to develop a product concept that addresses this need. This concept should be based on the information gathered during the market research and should be designed to meet the specific needs and preferences of the target market. The product concept should also be feasible, meaning that it can be developed and manufactured within the company's resources and capabilities. Once the product concept has been developed, the next step is to create a prototype of the product. This prototype is a physical model of the product that is used to test the concept and to gather feedback from potential customers. The prototype is typically made from a material that is easy to work with, such as wood or plastic, and is designed to look and function like the final product. Once the prototype has been created, the next step is to conduct a series of tests to evaluate the product's performance and to gather feedback from potential customers. These tests can be done in a variety of ways, including by having potential customers use the prototype and provide feedback, or by conducting laboratory tests to measure the product's performance. Once the tests have been completed, the next step is to refine the product concept based on the feedback received. This may involve making changes to the product's design, features, or materials. Once the product concept has been refined, the next step is to develop a detailed plan for the product's development and manufacturing. This plan should include information about the product's design, materials, and manufacturing process, as well as a timeline for the product's development and a budget for the project. Once the plan has been developed, the next step is to begin the development and manufacturing of the product. This involves working with a manufacturer to produce the product in a way that meets the company's quality standards and to ensure that the product is delivered to the target market in a timely and cost-effective manner. Finally, once the product has been developed and manufactured, the next step is to launch the product in the market. This involves creating a marketing plan that promotes the product and encourages potential customers to purchase it. The marketing plan should include information about the product's features and benefits, as well as a strategy for reaching the target market. Once the marketing plan has been developed, the next step is to launch the product in the market and to monitor its performance. This involves tracking sales, customer feedback, and other metrics to determine how well the product is performing in the market. If the product is not performing well, the company may need to make changes to the product or the marketing plan. If the product is performing well, the company may want to consider expanding the product line or launching new products in the future.



that one of the spark plug holes which provide access for the evacuation lines will be located as nearly at the bottom as possible. See paragraph 5.1 (c). The lines have been made as big as is consistent with required passage through the existing spark plug holes in the combustion chamber casing, to provide as much insurance as possible that available flow will exceed any leakage through the various joints and seals.

3.3 One of the major decisions as to how the test installation could best be made concerned the manner and location of the turbine in the wind tunnel system. It would have been the "cleanest" installation if the turbine and dynamometer could have been mounted as a unit entirely within the tunnel. This would have eliminated the need for a turbine shaft extension, since the dynamometer could have been coupled directly to the existing turbine shaft coupling. However, it is obvious that a dynamometer operating at the required speeds must depend on some sort of a fluid medium. The requirement of no contamination in the tunnel therefore ruled out this possibility, since the problem of perfectly sealing the dynamometer would surely have been a major stumbling block. The logical alternative was to locate at one of the elbows in the tunnel, so that the dynamometer, externally mounted, could be coupled to the turbine shaft extension projecting through the wall of the elbow. Rather than cut up the existing tunnel elbow at the desired location, it was decided to completely replace it with a new design. This gives the added advantage of designing for shortest possible shaft extension. The outermost passage diameter at inlet to the diffuser fortunately allowed direct mating to a 8-in. nominal-size pipe, and so, since the entire regular elbow required replacement, 8-inch pipe and fittings are used wherever





possible. Three elbows or types of turn were available, a standard 8-inch "short-radius" elbow, with an eight-inch turning radius on the centerline; a standard 8-in "long radius" elbow with twelve-inch turning radius on the centerline, and an assembly consisting of two sections of 8-in. tube joined at right angles by means of a section of turning vanes similar to the "Ducturn" presently used in the tunnel system. It was determined that the shortest possible shaft extension lengths were 14.915 in., 16.975 in., and 16.765 in. respectively. Since it was known that this length would be critical, it was decided to let it control the selection. The "short-radius" elbow is obviously the logical choice. Fig. 2 shows a development of the flow area which will be obtained through the "short-radius" elbow. It can be seen that pressure changes along the centerline will be regular and always in one direction, although highest acceleration occurs first on the under side of the nose cone, and subsequently on the upper side as the stream progresses through the elbow. This sequence may well be advantageous from the point of view of possible separation on the inside of the turn. At any rate, as has been mentioned previously, it is considered that any turbulence and other irregularities produced in the stream by the passage through the elbow will be obliterated sufficiently in the process of diffusion into the combustion chamber passages.

3.31 Appendix A presents an approximate analysis of the dynamics of the extension shaft required with the short-radius elbow. On the basis of an equivalent beam with pinned ends the critical speed would be 18,250 rpm. On the basis of the beam with fully fixed ends the critical speed would be 50,500 rpm. Although the maximum desired operating speed is almost identical with the former, the design is considered to be reasonably conservative in that the shaft extension is rigidly connected to other members at



[illegible]

both bearing supports, so that a considerable degree of fixity is obtained.

3.32 Appendix J shows that a tapered joint between the shaft extension and the dynamometer coupling is adequate for the applicable power transmission, so that costly splined joints can be eliminated.

3.33 The tunnel provides an airstream of controlled humidity. Further, the mass flow in the stream is normally measured at a point considerably downstream from the new elbow, and flow through the turbine must be known accurately. Therefore leakage into or out of the tunnel must be minimized, and for this reason the joint between the extension shaft and the tunnel elbow must be sealed as effectively as possible. This is accomplished by a labyrinth seal and gasket (Figs. 12 and 13). The seal is secured to the tunnel adapter pad (Fig. 11). Since the seal depends on very small clearances on the shaft for its effectiveness, and it is of significant length, the rigid specification that the inner face of the adapter pad be normal to the shaft axis, requiring final machining after assembly, is necessary. The securing bolt holes in the adapter have been specified oversize purposely to provide for proper radial positioning. Further, it was found that a simple enclosed radial ball bearing was adequate as a thrust bearing, but it is required that it be in very close alignment with the shaft. Hence the specification the adapter pad outer end be truly normal to the shaft axis. Although small discrepancies in axial position of the adapter face can be accommodated by shimming between the bearing spacer, (Fig. 14) and the adapter, such differences should be kept to a minimum, and therefore it has been specified that the outer face of the adapter also be machined after assembly into the elbow.



both parties reported, as that a considerable number of birds is present.

1.10. 1911. 4. There was a heavy rain between the two parties.

and the driver was unable to proceed the day before yesterday.

1.11. 1911. 5. The driver was unable to proceed the day before yesterday.

2.11. 1911. 6. The driver was unable to proceed the day before yesterday.

3.11. 1911. 7. The driver was unable to proceed the day before yesterday.

4.11. 1911. 8. The driver was unable to proceed the day before yesterday.

5.11. 1911. 9. The driver was unable to proceed the day before yesterday.

6.11. 1911. 10. The driver was unable to proceed the day before yesterday.

7.11. 1911. 11. The driver was unable to proceed the day before yesterday.

8.11. 1911. 12. The driver was unable to proceed the day before yesterday.

9.11. 1911. 13. The driver was unable to proceed the day before yesterday.

10.11. 1911. 14. The driver was unable to proceed the day before yesterday.

11.11. 1911. 15. The driver was unable to proceed the day before yesterday.

12.11. 1911. 16. The driver was unable to proceed the day before yesterday.

13.11. 1911. 17. The driver was unable to proceed the day before yesterday.

14.11. 1911. 18. The driver was unable to proceed the day before yesterday.

15.11. 1911. 19. The driver was unable to proceed the day before yesterday.

16.11. 1911. 20. The driver was unable to proceed the day before yesterday.

17.11. 1911. 21. The driver was unable to proceed the day before yesterday.

18.11. 1911. 22. The driver was unable to proceed the day before yesterday.

19.11. 1911. 23. The driver was unable to proceed the day before yesterday.

20.11. 1911. 24. The driver was unable to proceed the day before yesterday.

21.11. 1911. 25. The driver was unable to proceed the day before yesterday.

22.11. 1911. 26. The driver was unable to proceed the day before yesterday.

23.11. 1911. 27. The driver was unable to proceed the day before yesterday.

24.11. 1911. 28. The driver was unable to proceed the day before yesterday.

3.4 The low-speed branch of the wind tunnel structurally is essentially a large inverted "U" which is supported at its lower ends in the basement of the laboratory, with no support on the second floor where the test installation is to be made. Any elongation in the vertical section before the test installation must result in added displacement of the horizontal center line of the tunnel from the level of the second floor. This would preclude the possibility of mounting the dynamometer on the second floor independent of the tunnel structure, since it is imperative that the dynamometer and turbine shaft axes be in closest possible alignment due to the high operating speed. This may be accomplished most handily by mounting directly on the lower tunnel adapter flange at the top of the vertical section of the tunnel, securing with the flange bolts. This is shown schematically in Fig. 3. The dynamometer mount is essentially a table with angle-iron legs which straddle the 8-inch elbow and project down to and are secured to the adapter flange. The table must be so constructed that the dynamometer securing bolts may be located at will, to permit adjustment in angular alignment. Coincidence of the two shaft axes at the coupling is ensured by the coupling design.

3.41 The nicest installation would have provided a flexible and self-aligning coupling between the turbine and the dynamometer. Several Boston representatives of nationally-known makers of flexible couplings were contacted and queried about providing such couplings for this application. There were no such standard couplings usable at above 10,000 rpm., and reference to the engineering departments of the several companies yielded only negative results. All were reluctant to even advise relative to this





speed-power combination. The Fast's Coupling Department of the Koppers Company, Inc., offered a special coupling, their Dwg. No. CS-12520-4, at a cost of \$98.00. The coupling was 4 5/8-in. long and 4 15/16-in. on the O. D. The external area provided thereby was considered to be unacceptable due to the probability of showing high unmeasurable windage losses which would be a large portion of the total output of the turbine at low power-high speed settings. Therefore it was decided to specify a rigid coupling similar to the one used by the manufacturer of the turbo-jet engine. A rigid coupling is also specified for the junction between the turbine shaft and shaft extension. Twelve fitted coupling bolts are available from the old turbine-compressor connection. Due to the fact that this installation will be run at less than one-half the design peak torque, six of these available bolts will suffice in each of the two rigid couplings, and the coupling parts which are to be manufactured locally have been designed accordingly.

3.5 The turbine of the test installation is a high-speed device which is susceptible to very high acceleration in the unloaded condition. On several occasions when Westinghouse test personnel experienced no-load run-aways with a turbine of approximately the same size and power output, due to failure of water flow through their dynamometer, accelerations were estimated to be at the rate of 5000-10000 rpm/sec. It is obvious that an automatic system of emergency shut-down must be provided. It would be inadequate to provide merely for a tunnel compressor cut-off, since its inertia is considerable and there would also be a large mass of air under pressure upstream of the turbine. It was decided to provide





a safety valve in the air stream as close to the turbine as possible. This can be accomplished nicely at the lower tunnel adapter flange, and is represented by Fig. 5 and shown on Fig. 4. The spring loaded flapper valve is held open during normal operation by a latch. When the turbine exceeds a preset limit, the latch is withdrawn by a solenoid which is energized through a governor on the dynamometer. Ref. (a) shows schematically one possible design for the governor and associated electrical circuits. It should be modified to eliminate the emergency fuel cut-off (not needed) and to provide a tell-tale which would indicate that electrical energy is available for the solenoid circuit when needed. Provisions also should be made to cut off power to the wind tunnel compressor simultaneously with triggering of the safety valve. A hand hole is provided in the lower tunnel adapter flange to facilitate re-setting the safety valve with minimum alteration to the existing tunnel structure.

#### 4.0 INSTRUMENTATION

4.1 Static temperature and pressure must be known at any point where Reynolds' Number is to be computed, along with stream velocity. The latter can be determined by comparing stagnation and static pressures, both of which can be measured in an airstream with a sufficient degree of accuracy. However, static temperature is not measured readily with sufficient accuracy, and must be reduced from stagnation temperature by means of the relation

$$T(\text{stat.}) = T(\text{stag.}) \left( \frac{P(\text{stat.})}{P(\text{stag.})} \right)^{\frac{k-1}{k}}$$

assuming isentropic compression to stagnation conditions, where  $k$  is the ratio of the specific heats of the airstream. Stagnation temperature can





be measured with reasonable accuracy (ref. (e)). To summarize, stagnation temperature and stagnation and static pressure must be measured immediately ahead of the wheel, where Reynolds' Numbers are to be compared. Determination of stagnation temperature behind the turbine wheel together with stagnation temperature ahead, will provide a basis for computation of turbine output from the relation for an adiabatic process  $W_x = \Delta h = c_p (\Delta T_o)$ , where  $W_x$  is shaft work (see any reference on Flow Processes). This computation should be useful for comparison with output as measured by the dynamometer, therefore providing a basis for an estimation of combined losses between the turbine and the dynamometer, such as to bearings, seals, windage, etc. The various losses would not be separable. Measurement of either stagnation or static pressure behind the turbine will provide the necessary additional data for computation of work input to the turbines. Provisions have been made to measure these values (see Figs. 22 and 27), namely:

- (a) Total temperature at one location ahead of the turbine nozzles (at one side);
- (b) Total and static pressure at the same plane as above, at top and bottom of the flow annulus;
- (c) Total (and static, if required) pressure at three locations (top, side, and bottom) aft of the turbine, approximately  $3 \frac{1}{2}$  inches downstream in the straight portion formed by the tail cone of the turbo-jet engine. In this distance there will have been no change in the annular area or diameter from that of the turbine wheel, yet the flow will have been given a chance to obtain some degree of uniformity, and vibration-type impulses from the blades of the turbine will have been at least partially damped.

[illegible]



(d) Total temperature at the same plane as for (c), at a location half-way between two of the pressure measurement points.

4.11 The instrument access holes necessary for the above have all been dimensioned for an I. D. of  $5/8$ -in. so that an instrument head  $3/8$ -in. long by  $1/4$ -in. round or square can be inserted. Fig. 23 shows schematically the necessary mating bushings to support and provide a seal on the shanks of the instruments. These are designed to accommodate an O-ring packing to prevent leakage, yet permit angular and radial adjustment for traversing. The access provided will be sufficient to accommodate claw-type yaw meters or 3-hole yaw probes (ref. (c)), either of which could incorporate an additional tube for measurement of static pressure; Kiel-type total pressure probe heads (ref. (d)); or total temperature heads similar to some of those shown in ref. (e).

4.12 It is considered that only two measurements of total and static pressure ahead of the turbine should suffice, since there should be no swirl at that point, the only probable differences resulting from the short turning radius with possible separation in the 8-inch elbow ahead. Hence measurements of pressure are taken only at the top and bottom, between which the greatest differences should be apparent. Three points of pressure measurement aft of the turbine are provided, since it is believed that swirl components in the flow might make for non-uniform distribution, both radially and circumferentially. Further, measurements will be more difficult to make at this station if both total and static determinations are required, since the direction of flow must be found by yaw meter. It is suggested that only total pressure be measured at this location, using Kiel-type total pressure instruments which are largely insensitive to yaw. Since sufficient data



(4) Total percentage of the same class as for (3), as a function

of the difference between the two percentages mentioned above.

5.11 The function above is also a function for the other two cases

discussed in 5.10, i.e. for the case of a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.12 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.13 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.14 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.15 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.16 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.17 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.18 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.19 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.20 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.21 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.22 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.23 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.24 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.25 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.26 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.27 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.28 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.29 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.30 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.31 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.32 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.33 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.34 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

5.35 From the above we see that the function  $f$  is a function from  $\mathbb{R}^n$  to  $\mathbb{R}^n$ .

can be obtained from measurements ahead of the turbine to determine flow and Reynolds Number, static pressure behind would be redundant information.

4.13 The stagnation temperature in the tunnel is controllable, and the turbine inlet temperature should be maintained as close to ambient room temperature as possible, so that heat transfer to the surroundings from the stream will be minimized. Under these conditions it is believed that total temperature need only be measured at one circumferential location before and behind the turbine, since there is no apparent reason for circumferential variation at any given axial station.

4.14 No provisions have been made for external manipulation of the airflow instruments ahead of the turbine. It will be noted that the instrument lines are brought out through the turbine mounting flange, indicated on Fig. 3. It is predicted that it will suffice to mount the probes so that they will be on the mean radius of the turbine blades, and since essentially axial flow should be obtained at this point they could be so oriented and set. However, it is recommended that the pressure probes be of the yaw-meter type for use as a check on direction. If an error in orientation were to be made, the tunnel could be stopped and the probes re-oriented manually through the hand hold provided for this purpose in the turbine mounting flange. Once the proper orientation has been obtained it should not change for the various conditions of flow, since the volume flow will be essentially invariable. In the event that these provisions are proven to be inadequate, then additional access provisions can be made for this station similar to those let into the main tunnel wall in the transverse plane of the after-turbine station. The latter make it

1. The first step in the process of identifying a problem is to define the problem. This involves identifying the symptoms of the problem and determining the scope of the problem. Once the problem has been defined, the next step is to identify the causes of the problem. This involves identifying the factors that are contributing to the problem and determining the underlying causes. Once the causes have been identified, the next step is to develop a plan of action. This involves identifying the steps that need to be taken to solve the problem and determining the resources that will be needed to implement the plan. Finally, the last step in the process is to implement the plan and monitor the results. This involves putting the plan into action and tracking the progress of the solution. Once the problem has been solved, the final step is to evaluate the results and determine if the solution was effective. This involves comparing the results of the solution to the original problem and determining if the problem has been solved. If the problem has not been solved, the process may need to be repeated.

[illegible]



possible to manipulate the probes from outside the tunnel while running, since it would be beneficial to make radial traverses at the after station, and may prove to be profitable to determine stream directions at that station.

4.15 Torque and speed will be measurable at the dynamometer. A "strobo-tachometer" is available in the Gas Turbine Laboratory at M.I.T. which has been shown to give accuracy within 1 per cent, which is considered to be adequate for the intended use.

## 5.0 ASSEMBLY OF EQUIPMENT

5.1 When and as parts become available, the installation should be assembled in the general order following; (see ref. (h) for Westinghouse Parts identification.)

5.11 Install the after oil seal and gasket on the after end of the turbine bearings housing, Westinghouse Part No. 23F574-1. WARNING: Before this is done the turbine rotor and shaft can slide freely in the bearings housing, and careless handling may result in damaged bearings or rotor blades. It will be observed that production notes have been etched onto the turbine shaft at the place where the seal is to be effected. Any external projections of the upset metal should be carefully honed off before the seal installation. This honing should be kept to a minimum to prevent getting the shaft out of round at this point. Take all possible precautions to prevent grit or fragments from entering the bearings. The rubber gasket is brought into position from the shaft coupling end of the sub-assembly, requiring some stretching over projections in the process. (Do not attempt



to disassemble the turbine sub-assembly for gasket installation, due to the care required in re-assembly of shaft and coupling.) Then bring the sealing halves into position, locating the radial splits in the two pairs ninety degrees apart. Hold the sealing pieces very lightly in contact with the shaft while securing, to obtain minimum clearance with the shaft. The sealing crests have been made purposely sharp, to be worn off into clearance during the initial stages of running. Be sure that the gasket is centered on the shaft after final securing. Rotate the turbine wheel by hand to be sure too much interference has not been provided. Safety wire the securing screws.

5.12 Silver-solder the evacuation pipe adapter pads on to the turbine bearings support and drill and tap in place as shown on Fig. 17. Make a trial assembly of the evacuation pipes to be sure they do not project inside the I.D. of the bearings support. Provide for subsequent identification of mating parts. The pipes should be seven inches in length, of three-eighths inch standard pipe, belled at one end for clamped hose fittings, and having tapered pipe threads at the other end. Braze on the instrumentation access pads on the combustion chamber housing as shown in Fig. 22. Drill  $31/64$  inch hole through each and be sure that the hole is continuous into the airstream passage. Tap as specified on Fig. 22.

5.13 Ream out spark plug holes in the combustion chamber casing to an I.D. of  $13/16$  inch.

5.14 Install the diffuser tail cone and the combustion chamber inner liners (Figs. 15 and 16), ensuring that the various holes and cutouts



1. The first of these is the fact that the Commission has not yet received any information from the Government of the United States regarding the activities of the Committee for the Liberation of the People of the South (CLPS) in the United States.

Experiments were carried out with the following conditions: 1) The amount of water was 100 g; 2) The amount of water was 200 g; 3) The amount of water was 300 g; 4) The amount of water was 400 g; 5) The amount of water was 500 g; 6) The amount of water was 600 g; 7) The amount of water was 700 g; 8) The amount of water was 800 g; 9) The amount of water was 900 g; 10) The amount of water was 1000 g.

[illegible]

provided for the evacuation pipe installation are properly oriented. Then bring the combustion chamber housing and nozzle sub-assembly, and the diffuser and bearings support sub-assembly together, drawing up slowly with the securing screws, and secure. WARNING: The nozzle blades are very loosely fitted in their shrouds, and easily dropped out. Make sure that the nozzle inner shroud is brought into position on the bearings support end at the same rate the two sub-assemblies approach at the main securing flange. These sub-assemblies are indexed for proper orientation by two punch pricks on the mating flange of the combustion chamber housing and on one of the mounting lugs on the diffuser.

5.15 Secure the turbine shaft extension to the turbine, using six of the twelve bolts (WE Part.No. \*18H276 line 1) which effected the coupling between the turbine shaft and the compressor shaft. The resulting assembly should then be checked for static and dynamic balance, and run-out at the far end of the shaft extension. Safety with cotter keys, which should be weighed with their respective bolts before-hand. All possible care should be taken to ensure that weights are disposed symmetrically, for balance of a rapidly-rotating part which will be operating near its critical speed.

5.16 Assemble the Turbine Sub-assembly into the Bearing Support and Diffuser assembly. The after sealing gasket, Fig. 19, is mushroomed into the I.D. of the Bearings Support during this step, which will require some force. This force should not be imposed on the turbine or its shaft, since this would likely damage the outermost ridge of the labyrinth seal. All required force should be exerted by pulling on the bearings housing through





the diffuser. Safety-wire the securing bolts after drawing up tight.

5.17 Install the evacuation pipes, and pack in the spark-plug holes with cotton wicking, to minimize air leakage ahead of the turbine.

5.18 Install the nose cone (Fig. 21). Use gasket compound similar to Permatex, Type A, sparingly on as many of the mating surfaces as possible, taking care not to allow any to flow into the labyrinth sealing grooves or onto the turbine extension shaft. Remove all external excess compound after assembly. Rotate the turbine by hand to ensure that too much interference between labyrinth seals and shaft does not exist.

5.19 Slip the tunnel seal and gasket, Figs. 12 and 13, on to the turbine extension, but do not slide them too far along the shaft. Then mount the turbine test assembly on the turbine support flange and secure. Pull the tunnel seal and gasket onto the tunnel adapter pad (Fig. 11) from outside, and secure. The screw holes in the pad have been drilled oversize on purpose, to permit proper radial location of the seal and gasket relative to the shaft.

5.20 Install the thrust bearing on the dynamometer coupling half (Fig. 20), shrink on the bearing retainer ring (Fig. 26). Position spacer (Fig. 14).

5.21 Attach coupling-and-bearing assembly to the shaft extension and slip thrust bearing retainer (Fig. 14) into place, as shown in Fig. 3. Locate the centers of the two positioning pins from the bearing support. Remove support, coupling and bearing, and spacer, drill and tap for and install positioning pins.

5.22 Make final installation of the coupling assembly, torquing the

[illegible]

U.S. GOVERNMENT PRINTING OFFICE: 1967

Copyright © 2004 John Wiley & Sons, Ltd.

Further down the page, the text reads: "The following information is for your information only. It is not intended to be used as a substitute for professional advice. Please consult your physician or other health care provider for more information." This is a common disclaimer found in many health-related documents.

to make the first woman president.

After assembly, rotate the turbine to back at 1000 rpm for 10 min to

THESE TWO NEW FORMS ARE THE ONLY REPRODUCIBLE ONES

Noted on 11 Nov 1957, during the first survey at site 11.

*Journal of Management Education* 30(6) 789-804

IBM® Business Process Management Software is a registered trademark of International Business Machines Corporation.

Journal of the American Statistical Association

and reports. The report states that the data were obtained from the following sources:

[illegible]

retaining nut to 500 in-lbs. (It will be permissible to use a Stillson wrench on the stub end of the turbine rotor, but the resulting burrs should be dressed off carefully subsequently).

5.23 Feel out the resultant axial limits of the turbine rotor position (these will be small due to the presence of the after seal), and use split shims between the bearing spacer and the tunnel adapter pad to set the rotor half way between the limits. Rotate the turbine by hand to be sure that sufficient freedom exists for normal operation in the air stream.

5.24 Install the evacuation pipes and connect by hoses to a gas-tight access through the turbine mounting flange.

5.25 Install the tunnel safety valve on the lower tunnel adapter flange, as shown on Fig. 4. The safety valve springs are designed to be loaded by a turn of 70-deg. when the valve is in the closed position, and should be installed accordingly.

5.26 The remainder of the assembly procedure for getting the test installation into the tunnel is straightforward and may be seen clearly in Fig. 3.

## 6.0 OPERATING INSTRUCTIONS AND PRECAUTIONS

6.1 It is recommended that sand-bagging or equivalent protection be provided in the plane of the turbine wheel and that of the coupling between turbine shaft extension and dynamometer shaft before operation is attempted.





## 6.2 Starting

6.21 Calibrate and set the safety governor (Section 3.5) for activation at 20,000 rpm. Check the Primary Safety Mechanism. Be sure that the governor parts are working freely, without binding, and that the stand-by electrical circuit for the safety air valve tripping solenoid is energized (tell-tale light is on). Manipulate the governor so that the electrical circuit to the solenoid is closed, and be sure that the valve does in fact close. Reset, and close the hand-hole in the lower tunnel adapter flange.

6.22 Start oil-air mist evacuator, and establish pressure within the bearings support at a value lower than the lowest predicted static pressure at turbine inlet. Do not operate at any setting where this difference can not be maintained.

6.23 Start the oil pump for the oil air mist only after proper pressures have been set in accordance with 6.22. (This step might well be accomplished immediately after the turbine starts rotating, in step 6.24).

6.24 Set the water level control on the dynamometer for maximum torque (water chamber full).

6.25 Start the wind tunnel and establish desired flow.

6.26 Slowly reduce torque on the dynamometer to increase turbine speed to that desired. Watch carefully for vibrations.

6.27 Shut down tunnel immediately if safety solenoid circuit tell-tale light goes off, and re-establish the circuit before attempting further operation. It is to be strongly emphasized that this is a high-speed mechanism which is capable of very rapid acceleration on run-away due to torque failure.





### 6.3 Stopping

6.31 Increase dynamometer torque to maximum available.

6.32 Stop the wind tunnel, thereby stopping the turbine.

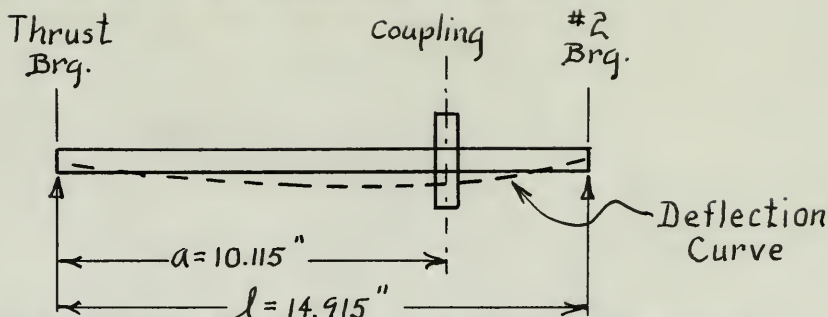
6.33 After the turbine has stopped, shut off the oil pump. (This might well be accomplished slightly before turbine stops in step 6.32).

6.34 Let the oil-air mist evacuator run as long as possible after stopping the pump, to provide assurance that all oil has been removed from the turbine and associated system. It must be emphasized that the interior of the wind tunnel system must not be contaminated in any way. Take all possible precautions to ensure that any leakage between the test assembly and the tunnel air stream is always inward through the various seals and toward the evacuation lines.



APPENDIX A

## DYNAMICS OF THE TURBINE SHAFT EXTENSION



The method of Rayleigh-Ritz for analysis of transversely vibrating beams will be followed. It will be remembered that, from the Law of Conservation of Energy, the maximum attainable potential energy of a vibrating system due to its position in space is equivalent to the maximum attainable kinetic energy of the system due to its motion, i.e.,

$$PE_{\max} = KE_{\max}$$

It will be assumed that the deflection curve for a vibrating pin-ended beam equivalent to the shaft system shown above can be represented by the equation

$$y = K \sin \left( \frac{\pi x}{\ell} \right) \cos (wt)$$

where  $K$  is some unknown constant

$w$  is rotational velocity, rad./unit time

Then the slope at any point is

$$\frac{dy}{dx} = K \frac{\pi}{\ell} \cos \left( \frac{\pi x}{\ell} \right) \cos (wt) = \theta$$

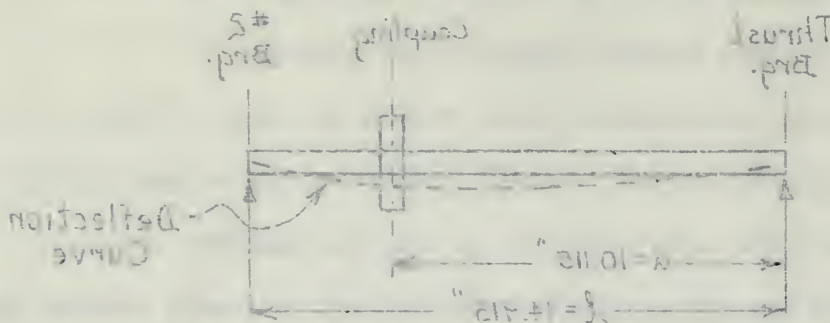
and curvature at any point is

$$\frac{d^2y}{dx^2} = -K \frac{\pi^2}{\ell^2} \sin \left( \frac{\pi x}{\ell} \right) \cos (wt) = \frac{d\theta}{dx}$$

The velocity at any point is

$$\frac{dy}{dt} = -K w \sin \left( \frac{\pi x}{\ell} \right) \sin (wt)$$





$$f(y) = \frac{1}{\sigma} \exp\left(-\frac{y}{\sigma}\right)$$

ed. 1989, pp. 12-13, 105-107.

$$Q = (m) \cos\left(\frac{x\pi}{a}\right) \cos \frac{\pi}{2} \quad Q = \frac{80}{20}$$

U.S. Department of Health and Human Services

$$\frac{d\lambda}{dt} = \lambda \left( \frac{1}{\lambda} \right) = \lambda \left( \frac{1}{\lambda} \right) = \frac{1}{\lambda} = \frac{1}{\lambda} = \frac{1}{\lambda}$$

2010-2011

$$\left( \frac{60}{5} \right) = 12 \text{ or } 12 = \frac{60}{5}$$

For a beam increment subjected to a bending moment  $M$ , which results in an angular deflection  $d\theta$ , the work done in bringing about the final condition of loading and position is

$$\Delta (PE) = M (\text{ave}) \times d\theta = 1/2 M d\theta$$

From Beam Theory we have

$$M = EI \frac{d^2 y}{dx^2} = EI \frac{d\theta}{dx}$$

Then

$$\Delta (PE) = 1/2 EI \frac{d\theta}{dx} d\theta = \frac{EI}{2} \left( \frac{d\theta}{dx} \right)^2 dx$$

and

$$\begin{aligned} PE_{\text{tot.}} &= \frac{EI}{2} \int_0^l \left( \frac{d\theta}{dx} \right)^2 dx = \frac{EI}{2} \int_0^l \left[ -K \frac{\pi^2}{l^2} \cos^2(\omega t) \sin^2\left(\frac{\pi x}{l}\right) \right]^2 dx \\ &= K^2 \frac{\pi^4 EI}{2l^4} \cos^2(\omega t) \int_0^l \sin^2\left(\frac{\pi x}{l}\right) dx \\ &= K^2 \frac{\pi^4 EI}{4l^3} \cos^2(\omega t) \end{aligned}$$

Incremental kinetic energy in the beam is  $\frac{dm}{2} \times \left( \frac{dy}{dx} \right)^2$

and that of the coupling is  $\frac{M_c}{2} \times \left( \frac{dy_c}{dt} \right)^2$

where  $M_c$  is the mass of the coupling.

but  $dm = \frac{\rho A dx}{g}$

where  $\rho$  is density,  $A$  is the sectional area

$$\begin{aligned} \text{Then } KE_{\text{tot}} &= K^2 \int_0^l \frac{\rho A}{2g} \omega^2 \sin^2(\omega t) \left[ \sin^2 \frac{\pi x}{l} \right] dx + \frac{M_c}{2} K^2 \omega^2 \sin^2(\omega t) \sin^2 \frac{\pi a}{l} \\ &= K^2 \left[ \frac{\rho A l}{4g} \omega^2 + \frac{M_c}{2} \omega^2 \sin^2 \left( \frac{\pi a}{l} \right) \right] \sin^2(\omega t) \end{aligned}$$

For a beam of length \$l\$, the total energy is \$E = \int\_0^l \frac{1}{2} \rho v^2 dx\$. The energy density is \$\frac{1}{2} \rho v^2\$. The energy flux is \$\frac{1}{2} \rho v^2 v\$. The energy flux is \$\frac{1}{2} \rho v^2 v\$.

$$\Delta \left( \frac{1}{2} \rho v^2 \right) = \frac{1}{2} \rho \frac{d}{dt} \left( \frac{1}{2} v^2 \right)$$

The total energy is \$E\$.

$$\frac{dE}{dt} = \frac{1}{2} \rho \frac{d}{dt} \left( \frac{1}{2} v^2 \right)$$

$$\Delta \left( \frac{1}{2} \rho v^2 \right) = \frac{1}{2} \rho \frac{d}{dt} \left( \frac{1}{2} v^2 \right)$$

$$\frac{d}{dt} \left[ \int_0^l \left( \frac{1}{2} \rho v^2 \right) dx \right] = \frac{1}{2} \rho \frac{d}{dt} \left( \frac{1}{2} v^2 \right)$$

$$\frac{d}{dt} \left[ \int_0^l \left( \frac{1}{2} \rho v^2 \right) dx \right] = \frac{1}{2} \rho \frac{d}{dt} \left( \frac{1}{2} v^2 \right)$$

$$\frac{d}{dt} \left[ \int_0^l \left( \frac{1}{2} \rho v^2 \right) dx \right] = \frac{1}{2} \rho \frac{d}{dt} \left( \frac{1}{2} v^2 \right)$$

$$\left( \frac{d}{dt} \right) \left[ \int_0^l \left( \frac{1}{2} \rho v^2 \right) dx \right] = \frac{1}{2} \rho \frac{d}{dt} \left( \frac{1}{2} v^2 \right)$$

$$\left( \frac{d}{dt} \right) \left[ \int_0^l \left( \frac{1}{2} \rho v^2 \right) dx \right] = \frac{1}{2} \rho \frac{d}{dt} \left( \frac{1}{2} v^2 \right)$$

The energy flux is \$\frac{1}{2} \rho v^2 v\$.

$$\frac{d}{dt} \left[ \int_0^l \left( \frac{1}{2} \rho v^2 \right) dx \right] = \frac{1}{2} \rho \frac{d}{dt} \left( \frac{1}{2} v^2 \right)$$

The energy flux is \$\frac{1}{2} \rho v^2 v\$.

$$\frac{d}{dt} \left[ \int_0^l \left( \frac{1}{2} \rho v^2 \right) dx \right] = \frac{1}{2} \rho \frac{d}{dt} \left( \frac{1}{2} v^2 \right)$$

$$\left( \frac{d}{dt} \right) \left[ \int_0^l \left( \frac{1}{2} \rho v^2 \right) dx \right] = \frac{1}{2} \rho \frac{d}{dt} \left( \frac{1}{2} v^2 \right)$$



Since

$$KE_{tot} = PE_{tot}$$

$$K^2 \left[ \frac{\rho A l}{4g} w^2 + \frac{M_c w^2}{2} \sin^2\left(\frac{\pi a}{l}\right) \right] \sin^2(wt) = K^2 \frac{\pi^4 EI}{4l^3} \cos^2(wt)$$

To maximize,  $\sin^2(wt) = \cos^2(wt) = 1$

(The maximum Potential Energy occurs at  $\frac{\pi}{2}$  radians before and after the time where Kinetic Energy is maximum)

$$\text{Therefore } w^2 \left[ \frac{\rho A l}{4g} + \frac{M_c}{2} \sin^2\left(\frac{\pi a}{l}\right) \right] = \frac{\pi^4 EI}{4l^3}$$

and

$$w = \sqrt{\frac{\pi^4 EI}{l^3} \times \frac{1}{\frac{\rho A l}{g} + 2M_c \sin^2\left(\frac{\pi a}{l}\right)}}$$

For the steel shaft of diameter 1 1/8 in.,

$$EI = 2.357 \times 10^6 \quad \text{lb.-sq.in.}$$

$$\rho = 0.286 \quad \text{lb./cu.in.}$$

$$a = 10.115 \quad \text{in.}$$

$$l = 14.915 \quad \text{in.}$$

$$g = 386 \quad \text{in./sec}^2$$

The coupling weight is 2.21 lb.

Solving the final equation above,  $w=1910$  rad./sec. which is, then, the critical speed for the shaft extension.

and

$$N = \frac{w}{2\pi} \times 60 = \frac{1905}{2\pi} \times 60 = 18,200 \text{ rpm}$$

For the case of the equivalent vibrating beam with rigidly fixed ends the

equation of the deflection curve is assumed to be

$$y = K \left[ 1 - \cos\left(\frac{2\pi x}{l}\right) \right] \cos(wt)$$

$$(f_w)^2 \cos \frac{\pi E I}{4 \lambda^4} K = (f_w)^2 \sin \left[ \left( \frac{\pi E I}{4 \lambda^4} \right) \sin \frac{\lambda^2 M}{2} + \frac{\lambda^2 A G}{2} \right]$$

To determine  $\sin(f_w) = \sin(f_w)$

(The constant  $\frac{\pi E I}{4 \lambda^4}$  is constant and  $\frac{\pi}{2}$ )

(The constant  $\frac{\pi E I}{4 \lambda^4}$  is constant)

$$\frac{\pi E I}{4 \lambda^4} = \left[ \left( \frac{\pi E I}{4 \lambda^4} \right) \sin \frac{\lambda^2 M}{2} + \frac{\lambda^2 A G}{2} \right]$$

$$W = \sqrt{\frac{\pi E I}{4 \lambda^4} \left[ \left( \frac{\pi E I}{4 \lambda^4} \right) \sin \frac{\lambda^2 M}{2} + \frac{\lambda^2 A G}{2} \right]}$$

The constant  $\frac{\pi E I}{4 \lambda^4}$  is constant

$$\frac{\pi E I}{4 \lambda^4} = \frac{\pi E I}{4 \lambda^4}$$

$$\frac{\pi E I}{4 \lambda^4} = \frac{\pi E I}{4 \lambda^4}$$

$$\frac{\pi E I}{4 \lambda^4} = \frac{\pi E I}{4 \lambda^4}$$

$$\frac{\pi E I}{4 \lambda^4} = \frac{\pi E I}{4 \lambda^4}$$

$$\frac{\pi E I}{4 \lambda^4} = \frac{\pi E I}{4 \lambda^4}$$

The constant  $\frac{\pi E I}{4 \lambda^4}$  is constant

Using the constant  $\frac{\pi E I}{4 \lambda^4}$  is constant

Using the constant  $\frac{\pi E I}{4 \lambda^4}$  is constant

$$\frac{\pi E I}{4 \lambda^4} = \frac{\pi E I}{4 \lambda^4}$$

The constant  $\frac{\pi E I}{4 \lambda^4}$  is constant

$$\frac{\pi E I}{4 \lambda^4} = \frac{\pi E I}{4 \lambda^4}$$

Proceeding as before  $\omega = 5280$  rad./sec.

and  $N = 50,500$  rpm.

These speeds represent the extremes between which it might be expected the shaft extension will have its critical speed. It is to be presumed that in the actual case there will be a considerable degree of fixity at both ends.



1. The first of these is the fact that the

the second is the fact that the

the third is the fact that the

the fourth is the fact that the

the fifth is the fact that the

the sixth is the fact that the

the seventh is the fact that the

the eighth is the fact that the

the ninth is the fact that the

the tenth is the fact that the

the eleventh is the fact that the

the twelfth is the fact that the

the thirteenth is the fact that the

the fourteenth is the fact that the

the fifteenth is the fact that the

the sixteenth is the fact that the

the seventeenth is the fact that the

the eighteenth is the fact that the

the nineteenth is the fact that the

the twentieth is the fact that the

the twenty-first is the fact that the

the twenty-second is the fact that the

APPENDIX B

## DETERMINATION OF TURBINE TEST OPERATING CONDITIONS

The well-known dimensionless parameters relating to speed and weight rate of flow in the turbine, for specification of performance, are:

$\frac{N}{\sqrt{\theta}}$  , where  $N$  = rpm.,  $\theta$  is ratio of actual temperature to standard temperature (usually 520 deg. R) in absolute units

$\frac{w_a \sqrt{\theta}}{\delta}$  , where  $w_a$  = weight rate of air flow (lbs./sec.)  
 $\theta$  is as above

$\delta$  is the ratio of actual static pressure to standard pressure (usually 14.7 psia or 29.92 in. Hg.)

Taking typical performance data from ref. (g) at rated power,

$$N = 34,000 \text{ rpm}$$

$$T_{t_1} = 1954 \text{ deg. R.} \quad \text{where } T_{t_1} \text{ is stagnation temperature ahead of the Turbine}$$

$$P_{t_1} = 84.63 \text{ in. Hg} \quad \text{where } P_{t_1} \text{ is stagnation pressure ahead.}$$

$$w_a = 6.64 \text{ lbs./sec.}$$

Tunnel Conditions for greatest weight flow can be maintained at 560 deg. R. and 1.6 atmospheres = 47.87 in.Hg., maximum desired.

Then, to obtain dynamic similarity between desired test conditions and design conditions

$$\left( \frac{N}{\sqrt{\theta}} \right)_{\text{Test}} = \left( \frac{N}{\sqrt{\theta}} \right)_{\text{Design}}$$

$$N_{\text{Test}} = N_{\text{Design}} \sqrt{\frac{\theta_{\text{Test}}}{\theta_{\text{Design}}}} = 34000 \sqrt{\frac{560}{1954}} = 18,220 \text{ rpm}$$

# APPENDIX B

## ANALYSIS OF THE DATA ON THE EFFECT OF TEMPERATURE ON THE RATE OF REACTION

The rate of reaction was measured at various temperatures and the results are given in Table I.

It is assumed that the reaction is first order with respect to the concentration of the reactant. The rate of reaction is given by the equation 
$$r = k[A]$$
 where  $k$  is the rate constant and  $[A]$  is the concentration of the reactant. The rate constant  $k$  is assumed to be independent of the concentration of the reactant.

The rate constant  $k$  is assumed to be independent of the concentration of the reactant. The rate constant  $k$  is assumed to be independent of the concentration of the reactant.

The rate constant  $k$  is assumed to be independent of the concentration of the reactant. The rate constant  $k$  is assumed to be independent of the concentration of the reactant.

The rate constant  $k$  is assumed to be independent of the concentration of the reactant. The rate constant  $k$  is assumed to be independent of the concentration of the reactant.

The rate constant  $k$  is assumed to be independent of the concentration of the reactant. The rate constant  $k$  is assumed to be independent of the concentration of the reactant.

The rate constant  $k$  is assumed to be independent of the concentration of the reactant. The rate constant  $k$  is assumed to be independent of the concentration of the reactant.

The rate constant  $k$  is assumed to be independent of the concentration of the reactant. The rate constant  $k$  is assumed to be independent of the concentration of the reactant.

The rate constant  $k$  is assumed to be independent of the concentration of the reactant. The rate constant  $k$  is assumed to be independent of the concentration of the reactant.

The rate constant  $k$  is assumed to be independent of the concentration of the reactant. The rate constant  $k$  is assumed to be independent of the concentration of the reactant.

The rate constant  $k$  is assumed to be independent of the concentration of the reactant. The rate constant  $k$  is assumed to be independent of the concentration of the reactant.

The rate constant  $k$  is assumed to be independent of the concentration of the reactant. The rate constant  $k$  is assumed to be independent of the concentration of the reactant.



$$\text{and } \left( \frac{w_a \sqrt{\theta}}{s} \right)_{\text{TEST}} = \left( \frac{w_a \sqrt{\theta}}{s} \right)_{\text{DESIGN}}$$

$$\text{so } w_{a_{\text{TEST}}} = w_{a_{\text{DESIGN}}} \times \frac{s_{\text{TEST}}}{s_{\text{DESIGN}}} \times \sqrt{\frac{\theta_{\text{DESIGN}}}{\theta_{\text{TEST}}}}$$

$$= 6.64 \times \frac{48.87}{84.63} \times \sqrt{\frac{1954}{560}}$$

$$w_{a_{\text{TEST}}} = 7.005 \text{ lbs./sec.}$$

$$\left( \frac{\sqrt{20}}{8} \right)_{\text{DESIGN}} = \left( \frac{\sqrt{20}}{8} \right)_{\text{TEST}}$$

$$\frac{\sqrt{20}}{8} \times \frac{1.27}{1.27} = \frac{\sqrt{20}}{8} \times \frac{1.27}{1.27}$$

$$\frac{\sqrt{20}}{8} \times \frac{1.27}{1.27} = \frac{\sqrt{20}}{8} \times \frac{1.27}{1.27}$$

$$W_{\text{TEST}} = 5.002 \text{ lb/sec}$$

APPENDIX C

## DETERMINATION OF STATE OF AIR AT TURBINE INLET

Area of the Turbine Annulus:

Outer Diameter = 8.1 in.

Inner Diameter = 5.1 in.

$$A = \frac{\pi}{4} \left( \frac{D_o^2 - D_i^2}{144} \right) = \frac{\pi}{4} \left[ \frac{(8.1)^2 - (5.1)^2}{144} \right] = 0.216 \text{ ft.}$$

Since only the stagnation conditions are pre-determined by the tunnel, static conditions must be obtained by trial and error. For first approximation use density based on stagnation conditions.

$$\rho_o = \frac{P_o}{RT_o} \quad \text{from Gas Laws, where the subscript "o" signifies stagnation conditions.}$$

$$= \frac{14.7 \times 1.6 \times 144}{53.35 \times 560} = 0.1132 \text{ lbs./cu.ft.}$$

$$\text{Then } V = \frac{w_a}{A\rho} \quad \text{where } V \text{ is stream velocity in ft./sec.}$$

$$= \frac{7.005}{.216 \times .1132} = 286 \text{ ft./sec.}$$

$$\begin{aligned} \text{Then } P &= P_o - \frac{\rho V^2}{2g} = 1.6 \times 14.7 \times 144 - 1/2 \times \frac{.1132}{32.2} \times (286)^2 \\ &= 3244 \text{ lbs./sq.ft.} \end{aligned}$$

$$\text{and } T = T_o - \frac{V^2}{2g J c_p} \quad \text{where the specific heat at constant pressure } C_p = 0.2399 \text{ BTU/#/deg. F at 560 deg. R.}$$

$$T = 560 - \frac{(286)^2}{64.4 \times 778 \times .2399} = 560 - 6.8 = 553.2$$



# APPENDIX I

MECHANICAL PROPERTIES OF THE POLYMER

Value of the constant  $k$

$$k = 1.1 \times 10^{-4}$$

$$k = 1.1 \times 10^{-4}$$

$$k = \frac{1}{4} \left( \frac{1}{1.1} - \frac{1}{1.1} \right) = 0.011 \text{ cm.}$$

From the experimental conditions the value of  $k$  is determined by the formula, which condition must be obtained in this and other. For this reason, the value of  $k$  is determined on experimental conditions.

$$k = \frac{1}{4} \left( \frac{1}{1.1} - \frac{1}{1.1} \right) = 0.011 \text{ cm.}$$

$$k = \frac{1.1 \times 10^{-4} \times 1.1}{1.1 \times 1.1} = 0.011 \text{ cm.}$$

$$k = \frac{1.1 \times 10^{-4} \times 1.1}{1.1 \times 1.1} = 0.011 \text{ cm.}$$

$$k = \frac{1.1 \times 10^{-4} \times 1.1}{1.1 \times 1.1} = 0.011 \text{ cm.}$$

$$k = \frac{1.1 \times 10^{-4} \times 1.1}{1.1 \times 1.1} = 0.011 \text{ cm.}$$

$$k = 0.011 \text{ cm.}$$

$$k = \frac{1.1 \times 10^{-4} \times 1.1}{1.1 \times 1.1} = 0.011 \text{ cm.}$$

$$k = \frac{1.1 \times 10^{-4} \times 1.1}{1.1 \times 1.1} = 0.011 \text{ cm.}$$

Then, for second trial,  $\rho = \frac{3244}{53.35 \times 553.2} = .1099 \text{ lbs./cu.ft.}$

$$V = \frac{7.005}{.216 \times .1099} = 295 \text{ ft./sec.}$$

$$P = 3241 \text{ lbs./sq.ft.} \quad \text{and } T = 553 \text{ deg. R}$$

The third trial shows

$$= .1098 \text{ lbs./cu.ft.} \quad V = 295 \text{ ft./sec.}$$

$$P = 3242 \text{ lbs./sq.ft.} = 22.5 \text{ lbs./sq.in.}$$

$$T = 553 \text{ deg. R}$$

From the second table,  $\frac{5000}{22.5 \times 22.5} = 9$ ,  $\frac{1000}{22.5 \times 22.5} = .4$

$$V = \frac{1000}{22.5 \times 22.5} = .4 \text{ sec.}$$

$$V = \frac{1000}{22.5 \times 22.5} = .4 \text{ sec.}$$

The third table shows

$$V = \frac{1000}{22.5 \times 22.5} = .4 \text{ sec.}$$

$$V = \frac{1000}{22.5 \times 22.5} = .4 \text{ sec.}$$

$$V = .4 \text{ sec.}$$



APPENDIX D

## DETERMINATION OF REYNOLDS' NUMBERS

Reynolds' Number for the required investigation will be based on state and velocity just ahead of the Turbine Nozzle Ring. The characteristic length will be the chord of the turbine blade at mean radius.

$$N_R = \frac{\rho V l}{\mu}, \text{ where } N_R \text{ is Reynolds' Number}$$

$\rho$  is mass density (lbs-sec<sup>2</sup>/ft<sup>4</sup>)

$V$  is velocity (ft/sec)

$l$  is characteristic length (ft)

$\mu$  is viscosity (lbs-sec/sq ft)

$$\mu = \mu_o \left( \frac{T_o + C}{T + C} \right) \left( \frac{T}{T_o} \right)^{1.5} \quad \text{from Ref. (i)}$$

where  $C = 120$  for air

$T_o = 273$  deg. C.

$\mu_o = 170.9 \times 10^{-6}$  poises

$T = 553 \times 5/9 = 308$  deg. C.

$$\mu = 170.9 \times 10^{-6} \left( \frac{273 + 120}{308 + 120} \right) \left( \frac{308}{273} \right)^{1.5} = 188.0 \times 10^{-6} \text{ poises}$$

$$= 188.0 \times 10^{-6} \times 2.088 \times 10^{-3} = 3.925 \times 10^{-7} \text{ lbs.-sec./sq.ft.}$$

$$l = \frac{1.1875}{12} = 0.099 \text{ ft.}$$

$$\text{Then } N_{R_{max}} = \frac{.1098 \times 295 \times 0.099}{32.2 \times 3.925 \times 10^{-7}} = 253,500.$$

The tunnel can be evacuated to one-tenth of the maximum pressure used above, at the same turbine inlet temperature

$$\text{Then } N_{R_{min}} = \frac{1}{10} \times N_{R_{max}} = 25,350$$

# APPENDIX I

## INVESTIGATION OF THE EFFECT OF TEMPERATURE ON THE RATE OF REACTION

The purpose of this investigation is to determine the effect of temperature on the rate of reaction between hydrogen peroxide and potassium iodide. The reaction is as follows:

$$2H_2O_2 \rightarrow 2H_2O + O_2$$

The rate of reaction is measured by the volume of oxygen gas evolved over a given period of time.

The following table shows the results of the experiment. The temperature was varied from 10°C to 30°C, and the volume of oxygen gas evolved was measured at regular intervals of time.

Temperature (°C)	Volume of $O_2$ (ml) at 10 min	Volume of $O_2$ (ml) at 20 min	Volume of $O_2$ (ml) at 30 min
10	10	20	30
20	20	40	60
30	30	60	90

$$k = \frac{1}{t} \ln \left( \frac{a}{a-x} \right) = \frac{1}{t} \ln \left( \frac{10}{10-10} \right) = \frac{1}{t} \ln \left( \frac{10}{0} \right)$$

where  $k$  is the rate constant,  $t$  is the time,  $a$  is the initial concentration, and  $x$  is the concentration of the reactant at time  $t$ .

$$\ln \left( \frac{a}{a-x} \right) = \ln \left( \frac{10}{10-10} \right) = \ln \left( \frac{10}{0} \right)$$

The rate of reaction is directly proportional to the concentration of the reactants.

$$k = \frac{1}{t} \ln \left( \frac{a}{a-x} \right)$$

$$\ln \left( \frac{a}{a-x} \right) = \ln \left( \frac{10}{10-10} \right) = \ln \left( \frac{10}{0} \right)$$

The results of the experiment show that the rate of reaction increases with increasing temperature.

$$k = \frac{1}{t} \ln \left( \frac{a}{a-x} \right)$$

APPENDIX E

## DETERMINATION OF TURBINE POWER OUTPUT

$$T_o = 560^\circ$$

$$P_o = 1.6 \text{ atmospheres} = 1.6 \times 14.7 = 23.53 \text{ psia.}$$

$$\text{Normal Rated Power is obtained at } \frac{P_i}{P_f} = 2$$

$$\begin{array}{l} \text{where } P_i \text{ is stagnation pressure at entrance} \\ P_f \text{ is stagnation pressure at exit} \end{array} \quad \left. \vphantom{\begin{array}{l} \text{where } P_i \text{ is stagnation pressure at entrance} \\ P_f \text{ is stagnation pressure at exit} \end{array}} \right\} \text{ see Ref. (g)}$$

$$\begin{array}{lll} \text{at } T_{oi} = 560 & H_i = 133.86 & P_{ri} = 1.5742 \\ & & \left. \vphantom{\begin{array}{l} P_{ri} = 1.5742 \\ h_f = 109.74 \end{array}} \right\} \text{ from Gas Tables.} \\ P_{rf} = \frac{1}{2} \times 1.5742 = .7871 & & h_f = 109.74 \end{array}$$

$$\Delta h_s = 133.86 - 109.74 = 24.12 \text{ BTU/lb.}$$

$$HP = \frac{24.12 \times 778}{550} W_a = 34.1 W_a \quad \text{assuming } \eta_t = 100$$

$$\begin{array}{l} \text{Therefore maximum possible power at desired operation} \\ = 34.1 \times 7.005 = 239.5 \text{ hp.} \end{array}$$



# APPENDIX A

## RELATIONSHIP OF PHYSICAL AND CHEMICAL PROPERTIES

$$T = 273 + t$$

$$V = 1.0 \text{ liter} = 1000 \text{ cm}^3 = 10^{-3} \text{ m}^3$$

$$S = \frac{1}{V} \left( \frac{\partial U}{\partial T} \right)_V$$

- (a) The total energy of the system is the sum of the internal energy and the work done by the system.

$$\begin{aligned} \left\{ \begin{aligned} \frac{\partial U}{\partial T} &= C_V \\ \frac{\partial U}{\partial V} &= P \end{aligned} \right. \quad \text{for an ideal gas} \end{aligned}$$

$$\Delta U = C_V \Delta T + P \Delta V$$

$$C_V = \frac{5}{2} R \quad \text{for a diatomic gas}$$

$$\Delta U = \frac{5}{2} R \Delta T + P \Delta V$$

$$\Delta U = \frac{5}{2} R \Delta T + P \Delta V$$

## APPENDIX F

## ANALYSIS OF THE CHARACTERISTICS OF A FLAT DISC RUNNING IN WATER

$$\text{H.P.} = \frac{4\pi f w^{n+1}}{(n+3) \times 550} R_0^{n+3} - R_1^{n+3} \quad (\text{see ref (f)})$$

where

$f$  = coefficient of friction

$w$  = angular velocity, radians/sec.

$R_0$  = outer radius of disc, ft.

$R_1$  = inner radius of water level, ft.

Experimental results show  $n = 1.8$ ,  $f = .004$  for rough surfaces,  $.002$  for polished surfaces.

$$R_0^{4.8} - R_1^{4.8} = \frac{\text{H.P.} \times 4.8 \times 550}{4\pi \times .004 \times w^{2.8}}$$

Design conditions are

$$N = 18,220 \text{ rpm}$$

$$\text{H.P.} = 239.5$$

Therefore it is desired to design a brake which will absorb 250 H.P. and over a range of  $\frac{3}{4}$  design speed to maximum allowable.

$$\frac{3}{4}N = 13,660 \text{ rpm}$$

$$\text{Maximum allowable} = 20,000 \text{ rpm}$$

It can be seen from the foregoing equation that the slowest speed will determine the largest radius. Therefore for 250 H.P. and 13,680 rpm

$$R_0^{4.8} - R_1^{4.8} = \frac{250 \times 4.8 \times 550}{4\pi \times .004 \times \left(\frac{2\pi}{60} \times 13,680\right)^{2.8}} = \frac{250 \times 4.8 \times 550}{4\pi \times .004 \times 6.82 \times 10^8}$$

$$R_0^{4.8} - R_1^{4.8} = .01929$$

# APPENDIX B

ANALYSIS OF THE OBSERVATIONS ON A SET OF DATA IN 1950

$$A.1.1. = \frac{1}{2} \left( \frac{1}{2} + \frac{1}{2} \right) = \frac{1}{2} \quad (see text (1))$$

where

$\frac{1}{2}$  = proportion of first

$\frac{1}{2}$  = regular values, regular

$\frac{1}{2}$  = value of first

$\frac{1}{2}$  = value of first

Experimental results:  $\frac{1}{2}$  = 1.1,  $\frac{1}{2}$  = 1.1, for each value, 1.00

for values of

$$\frac{1}{2} = \frac{1}{2} \left( \frac{1}{2} + \frac{1}{2} \right) = \frac{1}{2}$$

These conditions are

$\frac{1}{2}$  = 1.1, 1.1

$\frac{1}{2}$  = 1.1, 1.1

Therefore it is found that the values of  $\frac{1}{2}$  are 1.1, 1.1, and

over a range of  $\frac{1}{2}$  values, which is not negligible.

$$\frac{1}{2} = 1.1, 1.1$$

These values are 1.1, 1.1

It can be seen from the foregoing analysis that the values of  $\frac{1}{2}$  are

between the values 1.1, 1.1, and 1.1, 1.1

$$\frac{1}{2} = \frac{1}{2} \left( \frac{1}{2} + \frac{1}{2} \right) = \frac{1}{2}$$

$$\frac{1}{2} = \frac{1}{2} \left( \frac{1}{2} + \frac{1}{2} \right) = \frac{1}{2}$$



letting  $R_1 = 0$  for trial radius

$$R_0 = \sqrt[4.8]{.01929} = .439 \text{ ft.} = 5.26 \text{ in.}$$

Now set  $R_0 = 5.4 \text{ in.} = .45 \text{ ft.}$

$$R_1 = \sqrt[4.8]{(.45)^{4.8} - .01929} = \sqrt[4.8]{.0218 - .01929} = .288 \text{ ft.} = 3.46 \text{ in.}$$

for the lowest power output H.P. = 25

$$R_0^{4.8} - R_1^{4.8} = .001929$$

Under this condition the inner radius of the water level would be

$$R_1 = \sqrt[4.8]{.0218 - .0001929} = .441 \text{ ft.} = 5.292 \text{ in.}$$

For the maximum allowable speed and 250 H.P.

$$w = 2093$$

$$w^{2.8} = 1.991 \times 10^9$$

$$R_0^{4.8} - R_1^{4.8} = \frac{250 \times 4.8 \times 550}{4\pi \times .004 \times 1.991 \times 10^9} = .00659$$

$$R_1 = \sqrt[4.8]{.0218 - .00659} = .418 \text{ ft.} = 5.02 \text{ in.}$$

for lowest power output at 20,000 rpm

$$R_1 = \sqrt[4.8]{.0218 - .000659} = .4475 \text{ ft.} = 5.37 \text{ in.}$$

Compare this to the O.D. of the disc = 5.40. Instability is highly probable!!!

Water flow required for cooling purposes

Let allowable temperature rise = 100 F. degrees

$$\text{Heat in.} = \frac{\text{h.p.} \times 33000}{J} = \frac{250 \times 33000}{778} = 10,600 \text{ BTU/min.}$$

Let  $\theta = 0$  for trial solution

$$\text{and } \theta = 0 \Rightarrow \frac{1}{\rho} = \frac{1}{\rho_0} \sqrt{1 - \frac{8}{\rho_0^2}}$$

$$\text{and } \theta = 0 \Rightarrow \frac{1}{\rho} = \frac{1}{\rho_0} \sqrt{1 - \frac{8}{\rho_0^2}}$$

$$\text{and } \theta = 0 \Rightarrow \frac{1}{\rho} = \frac{1}{\rho_0} \sqrt{1 - \frac{8}{\rho_0^2}}$$

Let  $\theta = 0$  for trial solution

$$\frac{1}{\rho} = \frac{1}{\rho_0} \sqrt{1 - \frac{8}{\rho_0^2}}$$

Let  $\theta = 0$  for trial solution

$$\text{and } \theta = 0 \Rightarrow \frac{1}{\rho} = \frac{1}{\rho_0} \sqrt{1 - \frac{8}{\rho_0^2}}$$

Let  $\theta = 0$  for trial solution

$$\frac{1}{\rho} = \frac{1}{\rho_0} \sqrt{1 - \frac{8}{\rho_0^2}}$$

$$\frac{1}{\rho} = \frac{1}{\rho_0} \sqrt{1 - \frac{8}{\rho_0^2}}$$

$$\frac{1}{\rho} = \frac{1}{\rho_0} \sqrt{1 - \frac{8}{\rho_0^2}}$$

$$\text{and } \theta = 0 \Rightarrow \frac{1}{\rho} = \frac{1}{\rho_0} \sqrt{1 - \frac{8}{\rho_0^2}}$$

Let  $\theta = 0$  for trial solution

$$\frac{1}{\rho} = \frac{1}{\rho_0} \sqrt{1 - \frac{8}{\rho_0^2}}$$

Let  $\theta = 0$  for trial solution

Let  $\theta = 0$  for trial solution

Let  $\theta = 0$  for trial solution

$$\frac{1}{\rho} = \frac{1}{\rho_0} \sqrt{1 - \frac{8}{\rho_0^2}}$$

$$\text{wt.} = q/T = 10,600/100 = 106 \text{ lbs. of water/min.}$$

$$\text{gal./min.} = 106/8.345 = 12.71$$

It can be readily seen with this rate of flow and the narrow range of water level, that speed control would not be accurate enough for use.



...the ... of ...

...the ... of ...

...the ... of ...

...the ... of ...

...the ... of ...

...the ... of ...

...the ... of ...

...the ... of ...

...the ... of ...

...the ... of ...

...the ... of ...

...the ... of ...

...the ... of ...

...the ... of ...

...the ... of ...

...the ... of ...

...the ... of ...

...the ... of ...

...the ... of ...

APPENDIX G

## ANALYSIS OF THE CHARACTERISTICS OF A CYLINDER PARTIALLY SUBMERGED IN WATER

Assume that no end face is exposed to water

$$\text{Then } T = 2\pi f l w^n R^{n+2} \text{ (see Ref. (f))}$$

where  $T$  is torque on cylinder (ft-lb)

$f$  is coefficient of friction = 0.004

$l$  is length of the cylinder (ft)

$w$  is rotational speed, rad/sec

$R$  is the external radius (ft)

$$n = 1.8$$

$$\text{And Horsepower} = HP = T w / 550$$

$$= \frac{2\pi f l w^{n+1} R^{n+2}}{550}$$

$$\text{so } l = \frac{HP \times 550}{2\pi f w^{2.8} R^{3.8}} \quad \left( \begin{array}{l} \text{from Appendix F,} \\ w^{2.8} = 1.991 \times 10^9 \end{array} \right)$$

Then, for 250 hp. at 20,000 rpm,

$$l = 0.00275 \left( \frac{1}{R} \right)^{3.8}$$

Assume  $R = 2$  in.

$$\text{Then } l = .00275 \left( \frac{12}{2} \right)^{3.8} = 2.475 \text{ ft.} = 29.7 \text{ in.}$$

Power varies directly with length. For one-tenth of maximum desired operating output (=23.95 hp)

$$l = \frac{23.95}{250} \times 29.7 = 2.85 \text{ in.}$$


---

# APPENDIX B

ANALYSIS OF THE COMPARISON OF A SINGLE VARIABLE WITH A SINGLE

ANALYSIS OF THE COMPARISON OF A SINGLE VARIABLE WITH A SINGLE

ANALYSIS OF THE COMPARISON OF A SINGLE VARIABLE WITH A SINGLE

ANALYSIS OF THE COMPARISON OF A SINGLE VARIABLE WITH A SINGLE

ANALYSIS OF THE COMPARISON OF A SINGLE VARIABLE WITH A SINGLE

ANALYSIS OF THE COMPARISON OF A SINGLE VARIABLE WITH A SINGLE

ANALYSIS OF THE COMPARISON OF A SINGLE VARIABLE WITH A SINGLE

ANALYSIS OF THE COMPARISON OF A SINGLE VARIABLE WITH A SINGLE

ANALYSIS OF THE COMPARISON OF A SINGLE VARIABLE WITH A SINGLE

ANALYSIS OF THE COMPARISON OF A SINGLE VARIABLE WITH A SINGLE

$$\frac{100 \times 100}{100} = 100$$

$$\left( \frac{100 \times 100}{100} \right) = 100$$

$$\left( \frac{100 \times 100}{100} \right) = 100$$

ANALYSIS OF THE COMPARISON OF A SINGLE VARIABLE WITH A SINGLE

$$\left( \frac{100 \times 100}{100} \right) = 100$$

ANALYSIS OF THE COMPARISON OF A SINGLE VARIABLE WITH A SINGLE

$$\left( \frac{100 \times 100}{100} \right) = 100$$

ANALYSIS OF THE COMPARISON OF A SINGLE VARIABLE WITH A SINGLE

ANALYSIS OF THE COMPARISON OF A SINGLE VARIABLE WITH A SINGLE

$$\left( \frac{100 \times 100}{100} \right) = 100$$



Calculation of pressure forces required to balance centrifugal forces of the water rotating in the housing.

Centrifugal force of an element of water of unit depth

$$= \frac{\rho}{g} dr r d\theta \frac{V_r^2}{r}$$

where  $\rho$  = density (lbs./cu.ft.)

$g$  = gravitational constant (32.2 ft./sec.<sup>2</sup>)

$V_r$  = linear velocity at radius  $r$  (ft./sec.)

$r$  = radial position of the element (ft.)

$dr$  = radial length of the element

$rd\theta$  = the circumferential length

The balancing pressure force (centripetal force)

$$= \rho h \cdot rd\theta$$

where  $h$  is the hydrostatic head (ft.)

other symbols as above.

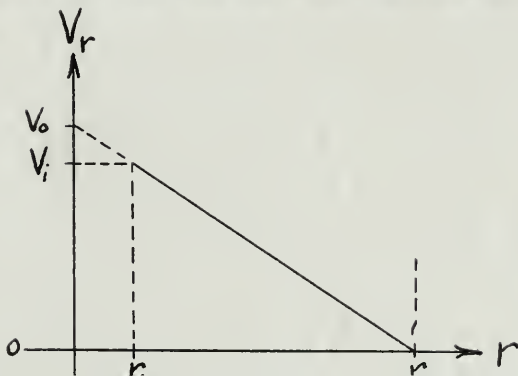
$$\text{Then } \rho h \cdot rd\theta = \frac{\rho}{g} \frac{V_r^2}{r} dr rd\theta$$

$$\text{and } h_{max} = \int_{r_i}^{r_c} \frac{1}{g} \frac{V_r^2}{r} dr$$

where  $r_c$  is the inner radius at the casing

$r_i$  is the outer radius of the cylinder.

Now assume velocity distribution from cylinder to casing is linear, coming to zero at the casing, and having zero slip on the cylinder, (shown herewith):



$V_i$  = velocity on surface of  
brake cylinder

$$= \omega r_i = \frac{20000}{60} \times 2\pi \times \frac{2}{12}$$

$$= 348 \text{ ft./sec.}$$

Calculation of average force required to remove residual forces

of the metal solution in the solution.

Consider the force of an element of width  $dx$  and

$$\frac{dF}{dx} = \frac{V}{L} \frac{dV}{dx}$$

where  $C$  = constant (100, 100, 100)

(5)  $V = V_0 + \frac{V_1 - V_0}{L} x$

$V_0$  = initial volume at  $x = 0$  (100, 100)

$V_1$  = initial volume at the element (100)

$L$  = initial length of the element

$V_0$  = the initial volume

The differential force (100, 100)

100, 100

where  $L$  is the initial length (100)

where  $L$  is the initial length

$$F = \frac{V}{L} \frac{dV}{dx} dx = \frac{V}{L} dV$$

$$F = \frac{V}{L} \frac{dV}{dx} dx = \frac{V}{L} dV$$

where  $L$  is the initial length

$V_0$  is the initial volume

For average volume (100, 100) the element is 100

where  $L$  is the initial length and initial force is 100, 100

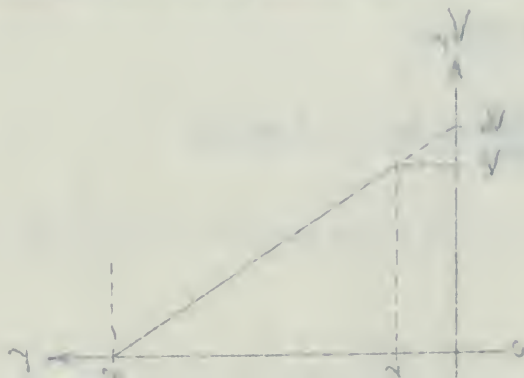
(100, 100)

$F = \frac{V}{L} \frac{dV}{dx} dx$

where  $L$  is the initial length

$$\frac{F}{L} = \frac{V}{L} \frac{dV}{dx} = \frac{V}{L} \frac{dV}{dx}$$

$V_0$  = initial volume



$$\text{Now } \frac{V_1}{r_c - r_1} = \frac{V_0}{r_c} \text{ and } V_0 = \frac{348 r_c}{r_c - \frac{2}{12}} = \frac{348 r_c}{r_c - .1667}$$

$$\text{and } \frac{r}{r_c} + \frac{V_r}{V_0} = 1$$

$$\text{so } V_r = \frac{r_c - r}{r_c} V_0 = \frac{348 (r_c - r)}{r_c - .1667}$$

$$\text{Then } h_{max} = \frac{348^2}{(r_c - .1667)^2} \times \frac{1}{g} \times \int_{.1667}^{r_c} \left( \frac{r_c^2}{r} - 2 r_c + r \right) dr$$

$$\text{so, when } r_c = 1 \text{ ft., } h_{max} = 3300 \text{ ft.}$$

$$\text{or } 1430 \text{ lbs./in.}^2$$

It can be seen from the nature of the derived formulation for  $h$  that it will decrease as  $r_c$  increases. However, if the enclosing casing is to be kept within reasonable dimensional limits, an enclosed pressure tight casing will be required, resulting in considerable unpredictable circulation. At low power settings with high speed there would be great likelihood of instability due to the short submerged depth on the brake cylinder. It would be impractical to decrease the cylinder diameter much under 4 in., because of the large submerged depth that would be required at full power.





APPENDIX HDETERMINATION OF POWER ABSORPTION IN A CENTRIFUGAL WATER PUMP

Assume that the water enters axially at the center and leaves tangentially at the periphery. Then power absorption is that due to the change in tangential momentum imposed on the water

$$HP = \frac{1}{2} \frac{WV^2}{33000} = \frac{1}{2} \frac{W}{g} \frac{V^2}{33000}$$

where  $W$  is the weight rate of water flow (lbs./min.)

$g$  is the gravitational constant (32.2 ft./sec.<sup>2</sup>)

$V$  is the linear velocity at the wheel rim (ft./sec.)

$$\text{and } W = \frac{2 \times HP \times g \times 33000}{V^2}$$

taking the density of water as 8 lbs./gal.

$$Q = \frac{2 \times HP \times g \times 33000}{8 \times V^2} = \frac{2 \times HP \times g \times 33000}{8 \times (wr)^2}$$

where  $Q$  is volume flow (gal./min.)

$w$  is the rotational speed (rad./sec.)

$r$  is the rim radius (ft.)

Then, for 250 hp. at 20000 rpm

$$Q = \frac{2 \times 250 \times 32.2 \times 33000}{8 \times \left(2\pi \times \frac{20000}{60}\right)^2 r^2} = 15.22 \times \frac{1}{r^2}$$

$$\text{also } V = wr = 2\pi \times \frac{20000}{60} \times r = 2092 r$$

The following solutions are found:

# CHAPTER II

THEORY OF THE EQUATION OF MOTION

Let us consider the motion of a particle in a medium.

Let  $x$  be the displacement of the particle from its equilibrium position.

Let  $m$  be the mass of the particle.

$$F = -kx$$

where  $k$  is the spring constant.

Let  $\omega$  be the angular frequency of the motion.

Let  $T$  be the period of the motion.

$$T = \frac{2\pi}{\omega}$$

Let  $\omega_0$  be the natural frequency of the system.

$$\omega_0 = \sqrt{\frac{k}{m}}$$

Let  $\omega$  be the driving frequency.

Let  $A$  be the amplitude of the motion.

Let  $\phi$  be the phase of the motion.

Let  $x(t)$  be the displacement as a function of time.

$$x(t) = A \cos(\omega t - \phi)$$

$$v(t) = -A\omega \sin(\omega t - \phi)$$

Let  $v(t)$  be the velocity as a function of time.



$r$ (ft.)	$wr = V$ (ft./sec.)	$Q$ (gal./min.)
0.5	1046	60.3
0.4	836	95.2
0.3	627	169.0

It can be seen that, for a reasonable flow, wheel diameters and resulting exit velocities are excessively high (see Section 3.13).

Year	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
1901	1.2	1.5	1.8	2.1	2.4	2.7	3.0	3.3	3.6	3.9	4.2	4.5
1902	1.3	1.6	1.9	2.2	2.5	2.8	3.1	3.4	3.7	4.0	4.3	4.6
1903	1.4	1.7	2.0	2.3	2.6	2.9	3.2	3.5	3.8	4.1	4.4	4.7

The following table shows the results of the experiments conducted during the year 1901. The results are given in the form of a table, and the data are taken from the original report.

The results of the experiments conducted during the year 1901 are given in the following table. The data are taken from the original report.

The results of the experiments conducted during the year 1901 are given in the following table. The data are taken from the original report.

The results of the experiments conducted during the year 1901 are given in the following table. The data are taken from the original report.

The results of the experiments conducted during the year 1901 are given in the following table. The data are taken from the original report.

The results of the experiments conducted during the year 1901 are given in the following table. The data are taken from the original report.

APPENDIX I

## THRUST BEARING DESIGN

Assume a 50% Reaction Turbine

$$\text{Then, when } \frac{(P_1)}{(P_f)_{\text{Design}}} = 2.$$

$$\frac{P_{1r}}{P_f} = \sqrt{2}$$

where  $P_1$  is stagnation pressure ahead of nozzles

$P_f$  is stagnation pressure behind rotor

$P_{1r}$  is stagnation pressure ahead of rotor

Maximum desired operating pressure  $P_1 = 1.6$  atmos.

$$\text{Then } P_{1r} = \frac{1.6 \times 14.7}{\sqrt{2}} = 16.62 \text{ psia}$$

$$P_f = \frac{1.6 \times 14.7}{2} = 11.78 \text{ psia}$$

$$\Delta p = 4.84 \text{ psia}$$

$$\text{Maximum Thrust} = A \times \Delta P$$

where  $A$  is turbine annulus area = .216 sq.ft.

$$T = .216 \times 144 \times 4.84 = 150.5 \text{ lbs.}$$

Consider Norma-Hoffman Double Plate Single Row Inch-Type Bearing

Medium size, Series 307-PP

Rated Load at 3000 rpm = 820 lbs.

Rated Load at 20000 rpm = 480 lbs., by extrapolation of tabulated values.

The manufacturer's catalog gives

$$P = 0.5 R + 1.5 T$$

Where  $P$  is equivalent load for rating

$R$  is radial Load

$T$  is Thrust Load



# PROBLEM 1

Given:  $\Delta x = 0.1$  m,  $\Delta t = 0.01$  s

Assume  $\Delta x$  and  $\Delta t$  are constant

$$\sqrt{s} = \frac{\Delta x}{\Delta t}$$

$$s = \frac{\Delta x}{\Delta t} = \frac{0.1}{0.01} = 10 \text{ m/s}$$

where  $\Delta x$  is the distance between two points

$\Delta t$  is the time interval between two points

$s$  is the speed of the object

where  $\Delta x = 0.1$  m,  $\Delta t = 0.01$  s

$$s = \frac{\Delta x}{\Delta t} = \frac{0.1}{0.01} = 10 \text{ m/s}$$

$$s = \frac{\Delta x}{\Delta t} = \frac{0.1}{0.01} = 10 \text{ m/s}$$

$$\Delta x = 0.1 \text{ m}$$

where  $\Delta x = 0.1$  m

where  $\Delta x = 0.1$  m,  $\Delta t = 0.01$  s

$$s = \frac{\Delta x}{\Delta t} = \frac{0.1}{0.01} = 10 \text{ m/s}$$

where  $\Delta x = 0.1$  m,  $\Delta t = 0.01$  s

where  $\Delta x = 0.1$  m,  $\Delta t = 0.01$  s

where  $\Delta x = 0.1$  m,  $\Delta t = 0.01$  s

where  $\Delta x = 0.1$  m,  $\Delta t = 0.01$  s

where  $\Delta x = 0.1$  m,  $\Delta t = 0.01$  s

$$s = \frac{\Delta x}{\Delta t} = \frac{0.1}{0.01} = 10 \text{ m/s}$$

where  $\Delta x = 0.1$  m,  $\Delta t = 0.01$  s

where  $\Delta x = 0.1$  m,  $\Delta t = 0.01$  s

where  $\Delta x = 0.1$  m,  $\Delta t = 0.01$  s

In the desired application  $R = 0$

$$\text{so } P = 1.5 \times 150.5 = 226 \text{ lbs.}$$

$$\text{Therefore Bearing factor of safety} = \frac{480}{226} = 2.2$$

(based on bearing life of 10000 hours, but extrapolating beyond maximum tabulated speed rating.)

#### Adequacy of Shrunk on Retainer Ring, Part No. D-1233.

When the tunnel is suddenly shut down, as when the air safety valve is triggered, the turbine becomes a compressor of low efficiency. This presumably happens when the load on the dynamometer is lost, and the turbine continues to run under its own inertia for a short period of time. Under these conditions the axial load on the turbine rotor reverses. Hence a shrunk on retainer ring will be required behind the bearing, on the coupling.

Use a steel ring with cross section  $1/8$  in. by  $1/4$  in. (see Fig. 26), with a Shrink Fit (Class 8) on the coupling where the O.D. is 1.3779 in.

Minimum interference is 0.0008 in., diametral

Now  $e = \frac{Pl}{AE}$ , where  $e$  = circumferential elongation in the ring

$P$  = the tensile load in the ring

$l$  = the mean circumference of the ring

$A$  is sectional area

$E$  is modulus of elasticity.

$= \pi(\Delta d)$ , where  $\Delta d$  is the interference

$$\text{Then } P = \frac{eAE}{l} = \frac{\pi \times \Delta d \times A \times E}{\pi \times d_m} = \frac{.0008 \times \frac{1}{4} \times \frac{1}{8} \times 30 \times 10^6}{3/2}$$

$$= 500 \text{ lbs.}$$





$$\text{Now } 2P = 2 p r w$$

where  $p$  is normal pressure on the inner surface of the ring

$r$  is the inner radius of the ring

$w$  is the axial width of the ring

$$\text{so } p = \frac{P}{rw}$$

$$\text{Total Normal Load} = N = p \times 2\pi r \times w = 2\pi P$$

$$N = 3140 \text{ lbs.}$$

$$\text{Allowable axial load} = N \times f$$

where  $f$  is static coef. of friction (0.15 for steel to steel)

$$\text{Allowable axial load} = 0.15 \times 3140 = 471 \text{ lbs.}$$

Assume that the turbine can provide a pressure ratio of 2/1 when acting as a pump (very conservative, since the blades will probably be stalled, and efficiency will be low).

Also assume that the tunnel compressor is working on a 2 to 1 pressure ratio, compressing to 1.6 atmospheres (the highest desired operating condition).

$$\text{Then exit pressure on turbine} = p_e = 1/2 \times 1.6 \times 14.7 = 0.8 \times 14.7$$

$$\text{and inlet pressure on turbine} = p_i = (1/2) p_e = 0.4 \times 14.7$$

$$\begin{aligned} \text{Then total axial pressure force on rotor} &= (0.8 - 0.4) \times 14.7 \times 144 \times .216 \\ &= 183 \text{ lbs.} \end{aligned}$$

Factor of Safety provided by shrink ring

$$= \frac{471}{183} = 2.58$$

For  $\alpha = 0.5$  and  $\beta = 0.5$

where  $\gamma$  is normal deviate on the lower tail of the t-distribution

$\gamma$  is the lower tail of the t-distribution

$\gamma$  is the lower tail of the t-distribution

$$\frac{1}{\sqrt{2\pi}} e^{-\frac{1}{2}\gamma^2}$$

Total normal deviate =  $\gamma + \frac{1}{\sqrt{2\pi}} e^{-\frac{1}{2}\gamma^2}$

where  $\gamma$  is normal deviate

where  $\gamma$  is normal deviate

where  $\gamma$  is normal deviate on the lower tail of the t-distribution

where  $\gamma$  is normal deviate on the lower tail of the t-distribution

where  $\gamma$  is normal deviate on the lower tail of the t-distribution

where  $\gamma$  is normal deviate on the lower tail of the t-distribution

where  $\gamma$  is normal deviate on the lower tail of the t-distribution

where  $\gamma$  is normal deviate on the lower tail of the t-distribution

where  $\gamma$  is normal deviate on the lower tail of the t-distribution

where  $\gamma$  is normal deviate on the lower tail of the t-distribution

where  $\gamma$  is normal deviate on the lower tail of the t-distribution

where  $\gamma$  is normal deviate on the lower tail of the t-distribution

where  $\gamma$  is normal deviate

where  $\gamma$  is normal deviate on the lower tail of the t-distribution

$$\frac{1}{\sqrt{2\pi}} e^{-\frac{1}{2}\gamma^2}$$

APPENDIX J

DETERMINATION OF TORQUE CAPACITY OF THE PLAIN TAPERED JOINT  
BETWEEN COUPLING AND SHAFT EXTENSION

Torque in Shaft =  $\frac{HP}{w}$  where  $HP$  is turbine power output  
 $w$  is turbine rotational speed

$$\text{Torque} = \frac{250 \times 33000}{20000 \times 2\pi} = 65.8 \text{ ft.-lbs.} = 789 \text{ in.-lbs.}$$

From ref.(I), static coef. friction, steel to steel, = 0.15

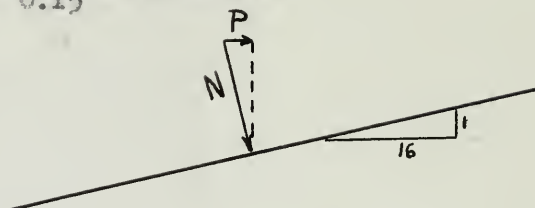
$$\text{Radial Taper in Joint} = 1/16 \text{ in/in}$$

$$\text{Average diam. in Joint} = 1 \frac{1}{16}$$

$$\text{Then total Surface force on Joint} = \frac{\text{Torque}}{\text{radius}}$$

$$= \frac{789}{17/32} = 1485 \text{ lbs.}$$

$$\text{and total Normal force} = N = \frac{1485}{f} = \frac{1485}{0.15} = 9,900 \text{ lbs.}$$



$$\frac{P}{N} = \frac{1}{\sqrt{1^2 + 16^2}} = \frac{1}{\sqrt{257}} = 0.0606$$

$$\text{so } P = 0.0606 \times 9,900 = 600 \text{ lbs.}$$

where  $P$  is the necessary axial force which must exist between coupling and shaft while running at full power.

The axial force on the turbine rotor tends to separate the two, so this also must be provided by the required retaining nut, i.e.

$$\text{Axial force imposed by retaining nut} = P = 600 + 150.5 = 750.5 \text{ lbs.}$$



# PROBLEM 1

Consider a particle of mass  $m$  moving in a circular path of radius  $R$ .

The particle is moving with a constant speed  $v$ .

Find the centripetal force  $F_c$  acting on the particle. (10 points)

$$F_c = \frac{mv^2}{R}$$

From part (1), the centripetal force is given by  $F_c = \frac{mv^2}{R}$ .

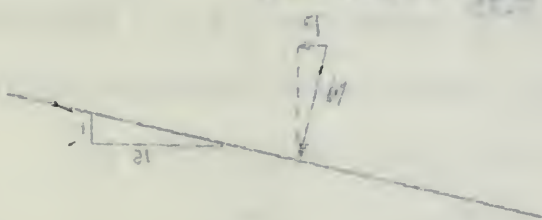
Substituting  $v = \frac{2\pi R}{T}$  into the equation for  $F_c$ , we get

$$F_c = \frac{m(2\pi R)^2}{RT^2}$$

$$F_c = \frac{4\pi^2 m R}{T^2}$$

$$F_c = \frac{4\pi^2 m R}{T^2}$$

$$F_c = \frac{4\pi^2 m R}{T^2}$$



$$F_c = \frac{mv^2}{R}$$

$$F_c = \frac{mv^2}{R}$$

From part (1), the centripetal force is given by  $F_c = \frac{mv^2}{R}$ .

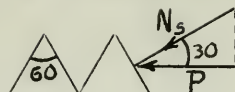
Substituting  $v = \frac{2\pi R}{T}$  into the equation for  $F_c$ , we get

$$F_c = \frac{4\pi^2 m R}{T^2}$$

Thus, the centripetal force is given by  $F_c = \frac{4\pi^2 m R}{T^2}$ .

$$F_c = \frac{4\pi^2 m R}{T^2}$$

Using a 5/8-in. screw, National Fine Thread Series (18thds./in.),



$$\text{Normal Force on Threads} = N_s = \frac{P}{\cos 30} = 866 \text{ lbs.}$$

$$\text{Torque on Threads} = N_s \times f \times r_t = T_t$$

where  $f$  is coef. of friction

$r_t$  is pitch radius of thread

Use  $f$  for static friction, to be conservative

$$\text{Then } T_t = 866 \times 0.15 \times \frac{0.59}{2} = 38.4 \text{ lb.-in.}$$

Assemble the coupling with 500 lb.-in. imposed on retaining nut.

$$\text{Factor of Safety at Full Load} = \frac{500}{38.4} = 13.0$$

It is recommended that high torque be used in setting up the assembly, to provide assurance that the coupling and shaft are properly aligned.



THE UNIVERSITY OF CHICAGO PRESS

$$u_1 = 0.19 \pm 0.01, \quad u_2 = 0.19 \pm 0.01, \quad u_3 = 0.19 \pm 0.01$$

It is recommended that this report be read in the following order:

...the ... ..



APPENDIX K

## AIR SAFETY VALVE SPRING DESIGN

Fig. 5 shows the desired valve.

O.D. = 9 in.                      Thickness = 1/4 in.

c.g. to hinge-line -- 5 in.

Material -- Aluminum Alloy, 24 ST

Density = 0.10 lbs./cu.in.

$$\begin{aligned} \text{Wt. of Valve} &= \frac{\pi}{4} \times D^2 \times t \times \rho = \frac{\pi}{4} \times 81 \times \frac{1}{4} \times 0.10 \\ &= 1.6 \text{ lbs.} \end{aligned}$$

Necessary seating torque about hinge = 1.6 x 4.62 = 7.72 lb.-in.

Use two springs, one left and one right hand, 0.093 in.diam. hard drawn spring wire on 0.625 mean spring diam., six active turns each.

$$\text{From ref. (j), } T = \frac{Ed^3n}{10.8 ND}$$

where E is modulus of Elasticity

d is spring wire diam.

n is angle of deflection (no. turns)

N is no. of active turns

D is mean diam. of the spring

T is the spring torque

Take spring deflection of 1/2 turn for valve-open position

$$\text{Then } T = \frac{30 \times 10^6 \times (.093)^4 \times 1/2}{10.8 \times 6 \times 0.625} = 27.8 \text{ lb.-in. each}$$

# APPENDIX A

THEORY OF THE EARTH'S MAGNETISM

Let  $\theta$  be the angle between the axis of the Earth and the axis of the magnetic field.

$$\cos \theta = \frac{1}{2} \quad \text{or} \quad \theta = 60^\circ$$

$$\sin \theta = \frac{\sqrt{3}}{2} \quad \text{or} \quad \theta = 60^\circ$$

$$\tan \theta = \frac{\sqrt{3}}{1} \quad \text{or} \quad \theta = 60^\circ$$

$$\sec \theta = \frac{2}{1} \quad \text{or} \quad \theta = 60^\circ$$

$$\csc \theta = \frac{2}{\sqrt{3}} \quad \text{or} \quad \theta = 60^\circ$$

$$\cot \theta = \frac{1}{\sqrt{3}} \quad \text{or} \quad \theta = 60^\circ$$

$$\operatorname{cosec} \theta = \frac{2}{\sqrt{3}} \quad \text{or} \quad \theta = 60^\circ$$

$$\sec \theta = \frac{2}{1} \quad \text{or} \quad \theta = 60^\circ$$

$$\tan \theta = \frac{\sqrt{3}}{1} \quad \text{or} \quad \theta = 60^\circ$$

$$\cot \theta = \frac{1}{\sqrt{3}} \quad \text{or} \quad \theta = 60^\circ$$

$$\operatorname{cosec} \theta = \frac{2}{\sqrt{3}} \quad \text{or} \quad \theta = 60^\circ$$

$$\sec \theta = \frac{2}{1} \quad \text{or} \quad \theta = 60^\circ$$

$$\tan \theta = \frac{\sqrt{3}}{1} \quad \text{or} \quad \theta = 60^\circ$$

$$\cot \theta = \frac{1}{\sqrt{3}} \quad \text{or} \quad \theta = 60^\circ$$

$$\operatorname{cosec} \theta = \frac{2}{\sqrt{3}} \quad \text{or} \quad \theta = 60^\circ$$

$$\sec \theta = \frac{2}{1} \quad \text{or} \quad \theta = 60^\circ$$

$$\tan \theta = \frac{\sqrt{3}}{1} \quad \text{or} \quad \theta = 60^\circ$$

$$\cot \theta = \frac{1}{\sqrt{3}} \quad \text{or} \quad \theta = 60^\circ$$

Angle of turn when valve is seated is 70 deg.

$$\text{Then } T_{\text{seat.}} = \frac{70}{180} \times 27.3 \times 2 = 21.6 \text{ in lbs. seating torque}$$

Factor of Safety when seated

$$= \frac{21.6}{7.72} = 2.8, \text{ and valve will stay safely closed without a latch.}$$

See Fig. 7 for spring details

Analysis of Closing Time of Air safety valve:

$$\alpha = \frac{d^2\theta}{dt^2} = \frac{T_\theta}{I}$$

where  $\alpha$  = angular acceleration

$\theta$  = angle of valve, referenced to zero at open position (radians)

$T_\theta$  = torque on the valve, a function of  $\theta$

$I$  = Mass moment of Inertia of the valve about the hinge.

When valve is open,  $T_o = 55.6 \text{ lb.-in.}$

When valve is closed,  $T_c = 21.6 - 7.72$   
 $= 13.88 \text{ lb. in.}$

$$\frac{\theta_T}{55.6} = \frac{110/57.3}{55.6 - 13.88}, \text{ so } \theta_T = \frac{55.6}{41.72} \times \frac{110}{57.3} = 2.56 \text{ rad.}$$

$$\text{Then } T_\theta = 55.6 \left(1 - \frac{\theta}{\theta_T}\right) = 55.6 - 21.7 \theta$$

$$\text{so } \frac{d^2\theta}{dt^2} = \frac{55.6 - 21.7 \theta}{I}$$

$$\text{and } \frac{d^2\theta}{dt^2} + \frac{21.7}{I} \theta = \frac{55.6}{I}$$

angle of view was 15 degrees at 100 ft.

$$\text{Angle of view} = \frac{100}{100} = 1 \text{ radian}$$

Angle of view was 15 degrees

$$\text{Angle of view} = \frac{15}{180} \times \pi = 0.26 \text{ radian}$$

Angle

Angle of view was 15 degrees

-----

Angle of view was 15 degrees

$$\frac{15}{180} \times \pi = 0.26$$

Angle of view was 15 degrees

Angle of view was 15 degrees

Angle of view was 15 degrees

Angle of view was 15 degrees

Angle of view was 15 degrees

Angle of view was 15 degrees

Angle of view was 15 degrees

$$\text{Angle of view} = \frac{15}{180} \times \pi = 0.26$$

$$\text{Angle of view} = \frac{15}{180} \times \pi = 0.26$$

$$\text{Angle of view} = \frac{15}{180} \times \pi = 0.26$$

$$\text{Angle of view} = \frac{15}{180} \times \pi = 0.26$$



One possible solution is

$$\theta = C_1 \cos \sqrt{\frac{21.7}{I}} t + C_2 \sin \sqrt{\frac{21.7}{I}} t + \frac{55.6}{21.7}$$

$$t = 0 \text{ when } \theta = 0 \quad \text{and } w = \frac{d\theta}{dt} \text{ when } \theta = 0$$

$$\text{Then } C_1 = -\frac{55.6}{21.7} \quad \text{and } C_2 = 0$$

$$\text{so } \theta = \frac{55.6}{21.7} \left[ 1 - \cos \sqrt{\frac{21.7}{I}} t \right]$$

$$\text{When the valve is closed, } \theta_c = \frac{110}{57.3} \text{ rad.}$$

$$\text{so } \frac{110}{57.3} = \frac{55.6}{21.7} \left[ 1 - \cos \sqrt{\frac{21.7}{I}} t \right]$$

$$1 - \cos \sqrt{\frac{21.7}{I}} t = .750$$

$$\text{Now } I = \frac{W}{g} \left( \frac{D^2}{16} + d^2 \right)$$

where  $W$  is weight of the valve in lbs.

$g$  is gravitational constant (386 in./sec<sup>2</sup>)

$D$  is the valve diameter (9 in.)

$d$  is distance from the c.g. of the valve to the hinge line (4.82 in.)

$$I = \frac{1.6}{386} \left( \frac{81}{16} + 23.25 \right) = .1172 \text{ lbs.-in.-sec.}^2$$

$$\cos \sqrt{\frac{21.7}{.1172}} = \cos \sqrt{185} = \cos (13.6 \text{ rad.}) = .515$$

$$\text{so the required time for closing} = t = \frac{1 - .750}{.515}$$

$$t = 0.485 \text{ sec.}$$

It is to be noted that this represents the minimum closing time, since the air stream will provide considerable additional closing effect.

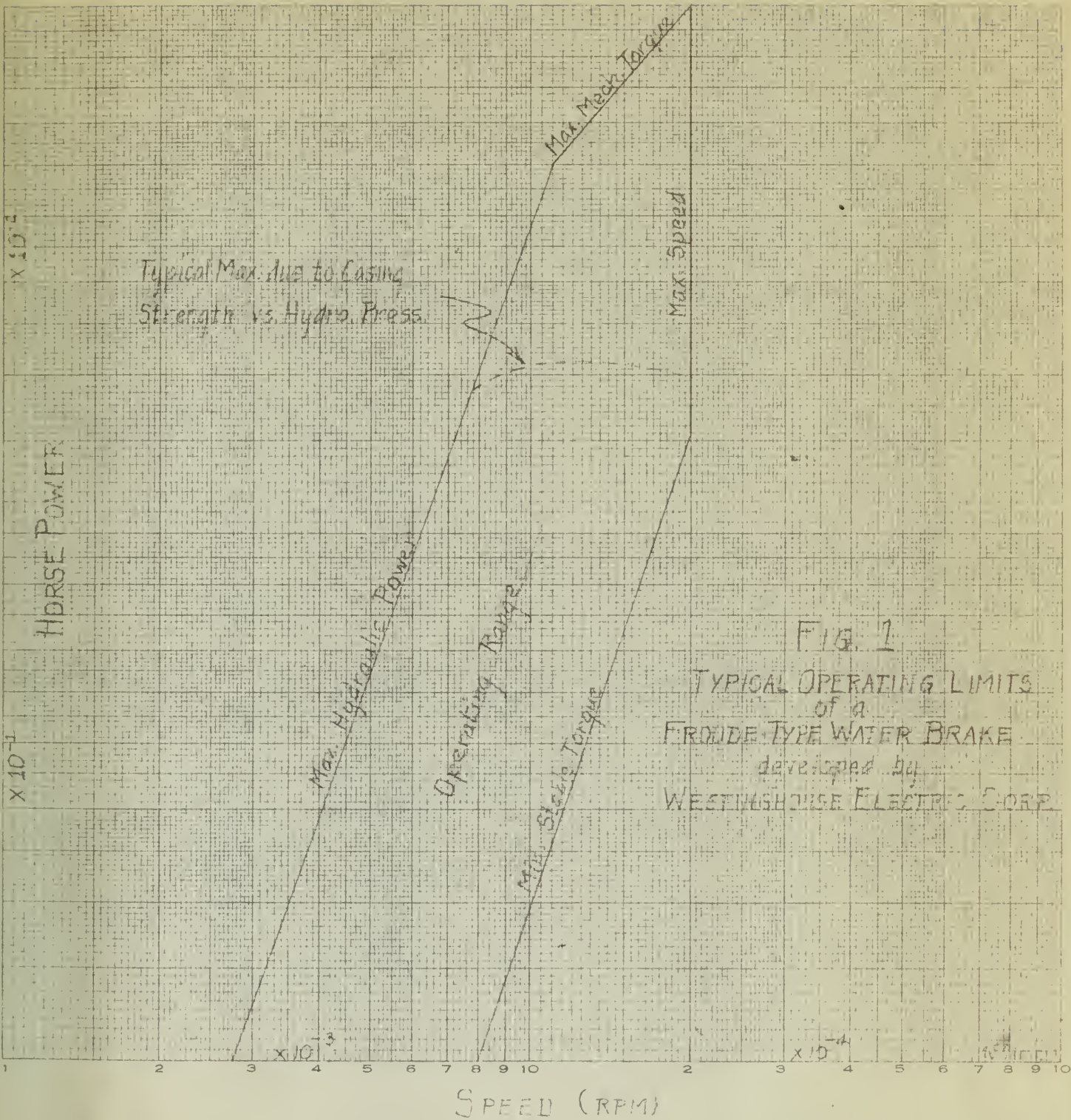


## References:

- (a) "The Performance of Axial-Flow Turbines", D. G. Ainley, B.Sc., The Institution of Mechanical Engineers, Proceedings 1948, Vol. 159, War Emergency Issue No. 41, London.
- (b) "An Investigation of the Properties of Eddy Current Brakes", Lof, J.L.C., Master's Thesis 1941, MIT.
- (c) "Report on Characteristics of a 3 Hole Yaw Probe", Vance D. Baker, July 10, 1946, Pratt and Whitney Aircraft Division of the United Aircraft Corporation. (Available in the Gas Turbine Library, MIT).
- (d) "Report on Kiel Probes", Marston Moffatt, October 17, 1945, Pratt and Whitney Aircraft Division of the United Aircraft Corporation. (Available in the Gas Turbine Library, MIT).
- (e) "Report on Cylindrical Temperature Probes", Marston Moffatt, May 18, 1945, Pratt and Whitney Aircraft Division of the United Aircraft Corporation. (Available in the Gas Turbine Library, MIT).
- (f) "Hydraulics and Its Applications", A.H. Gibson, Third Edition, 1925, D. Van Nostrand Company, New York, N.Y.
- (g) "Turbo-Jet Aviation Log Book, Westinghouse Engine No. 5A1314-10". (In custody of the Gas Turbine Library, MIT).
- (h) "Handbook of Service Instructions for Turbo-Jet Engines," Model XJ32-WB-4 (X9.5B). (In custody of Gas Turbine Library, MIT).
- (i) "Handbook of Engineering Fundamentals" O. W. Eshbach, 11th Printing, 1947, John Wiley & Sons, Inc. New York, N. Y.
- (j) "Mechanical Spring Design Catalog", Wallace Barnes Co., Bristol, Conn.

1. "The Investigation of the Committee of the United States Senate on the Activities of the Communist Party in the United States, 1947-1954." (Available in the New York Public Library, New York, N.Y.)
2. "The Investigation of the Committee of the United States Senate on the Activities of the Communist Party in the United States, 1947-1954." (Available in the New York Public Library, New York, N.Y.)
3. "The Investigation of the Committee of the United States Senate on the Activities of the Communist Party in the United States, 1947-1954." (Available in the New York Public Library, New York, N.Y.)
4. "The Investigation of the Committee of the United States Senate on the Activities of the Communist Party in the United States, 1947-1954." (Available in the New York Public Library, New York, N.Y.)
5. "The Investigation of the Committee of the United States Senate on the Activities of the Communist Party in the United States, 1947-1954." (Available in the New York Public Library, New York, N.Y.)
6. "The Investigation of the Committee of the United States Senate on the Activities of the Communist Party in the United States, 1947-1954." (Available in the New York Public Library, New York, N.Y.)
7. "The Investigation of the Committee of the United States Senate on the Activities of the Communist Party in the United States, 1947-1954." (Available in the New York Public Library, New York, N.Y.)
8. "The Investigation of the Committee of the United States Senate on the Activities of the Communist Party in the United States, 1947-1954." (Available in the New York Public Library, New York, N.Y.)
9. "The Investigation of the Committee of the United States Senate on the Activities of the Communist Party in the United States, 1947-1954." (Available in the New York Public Library, New York, N.Y.)
10. "The Investigation of the Committee of the United States Senate on the Activities of the Communist Party in the United States, 1947-1954." (Available in the New York Public Library, New York, N.Y.)
11. "The Investigation of the Committee of the United States Senate on the Activities of the Communist Party in the United States, 1947-1954." (Available in the New York Public Library, New York, N.Y.)
12. "The Investigation of the Committee of the United States Senate on the Activities of the Communist Party in the United States, 1947-1954." (Available in the New York Public Library, New York, N.Y.)







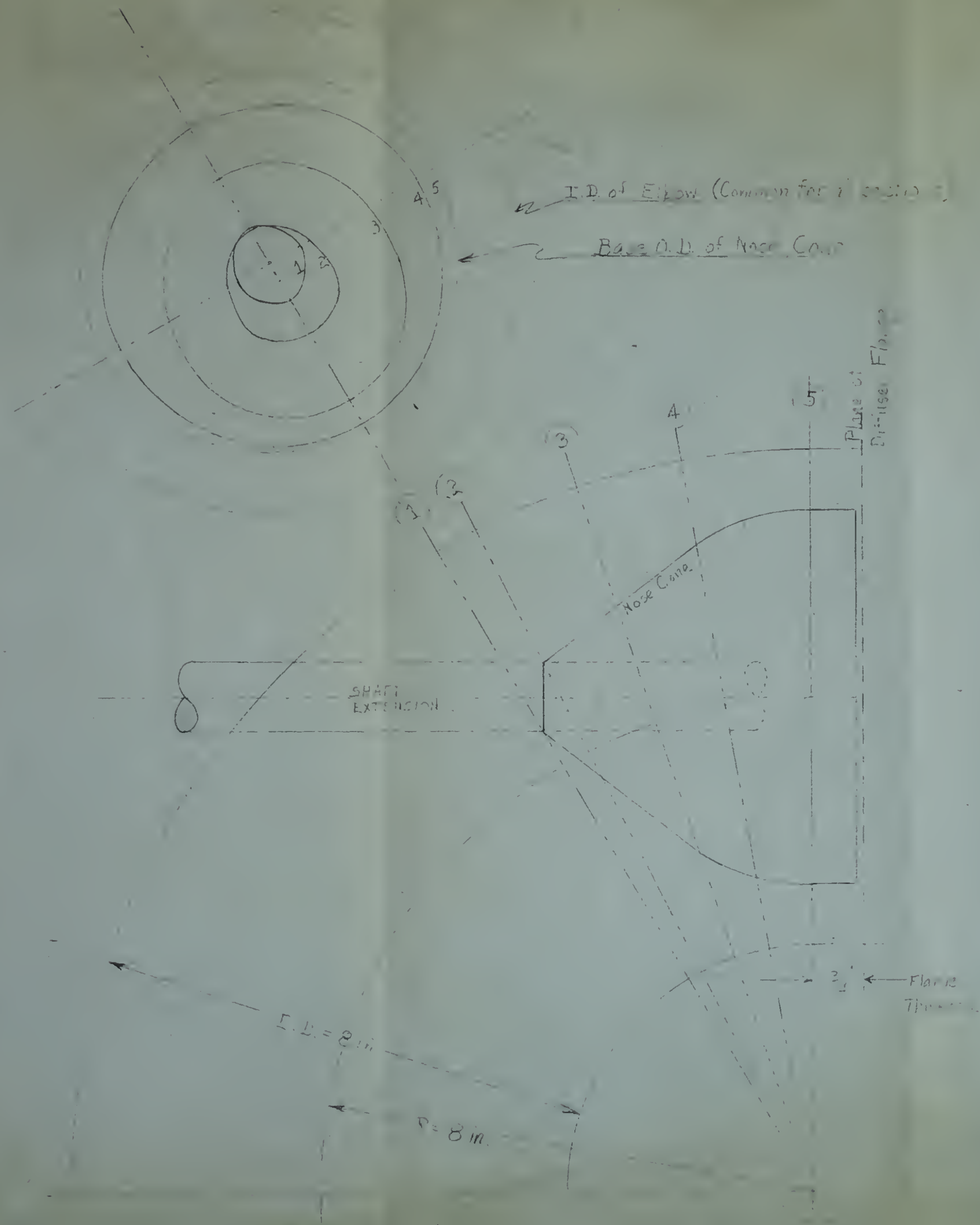


FIG. 2  
Sectional Shapes of Stream Passages  
Through 8 in. "Short Radius" Elbow

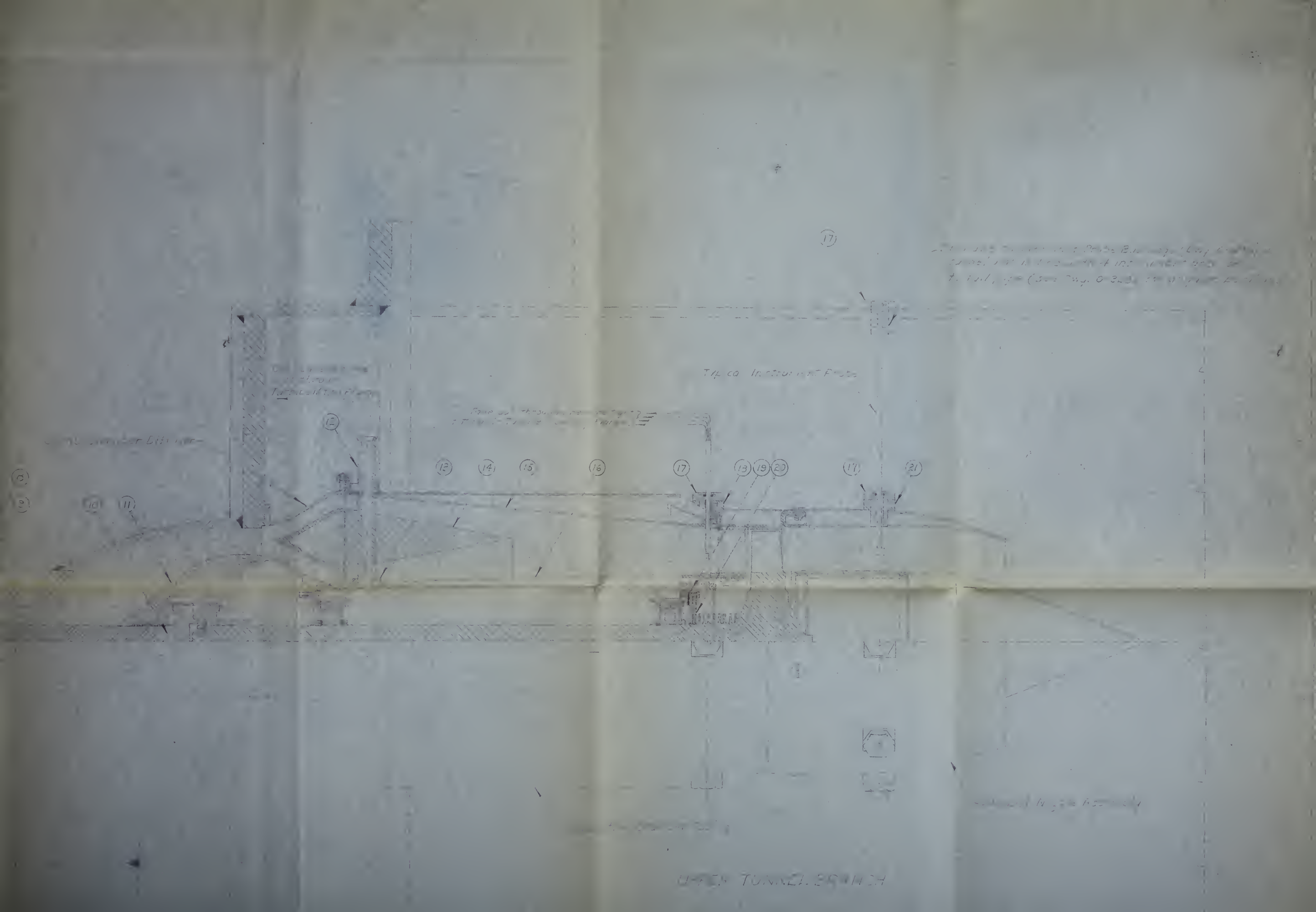


BILL OF MATERIAL	
1	CONCRETE PUMP AND 50' PIPE
2	CONCRETE PUMP
3	CONCRETE PUMP
4	CONCRETE PUMP
5	CONCRETE PUMP
6	CONCRETE PUMP
7	CONCRETE PUMP
8	CONCRETE PUMP
9	CONCRETE PUMP
10	CONCRETE PUMP
11	CONCRETE PUMP
12	CONCRETE PUMP
13	CONCRETE PUMP
14	CONCRETE PUMP
15	CONCRETE PUMP
16	CONCRETE PUMP
17	CONCRETE PUMP
18	CONCRETE PUMP
19	CONCRETE PUMP
20	CONCRETE PUMP
21	CONCRETE PUMP

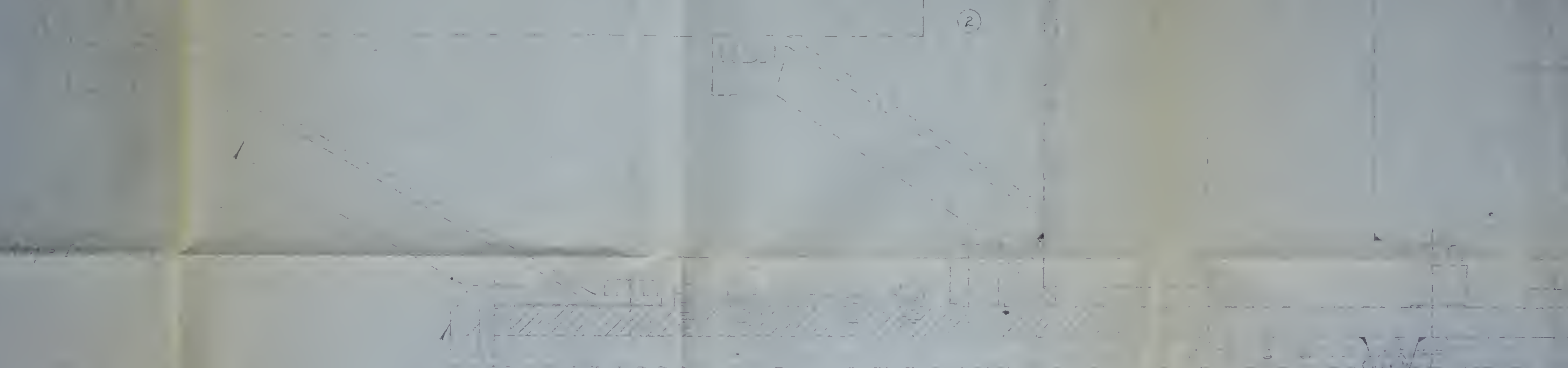
DYNAMOMETER







LYNCHMETER



①

LOWER TUNNEL BRANCH

100  
100  
100



UPPER TUNNEL BRANCH

NO.	MATERIAL	NO. 710
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GAS TURBINE LABORATORY		
GAS TURBINE TEST INSTAL.		
GENERAL ASSEMBLY		
DRAWN BY	DATE 5/13/49	
CHECKED BY	SCALE Half size	
DATE 5/13/49		0-5259

FIG. 3

13 1/2" 20.1" dia

Drill 3/8" holes 3/4" apart

5" dia  
hand hole

5 1/2" dia  
Circle

6"

4 1/2"

Drill 1/2" holes 1" apart



# BILL OF MATERIAL

	PART NO.	NAME	QTY.
1	O-4260	Flange	1
2	O-1265	Hard Rock Cover	1
3		1/8" x 1/8" x 1/8" Grommet	1
4	3" x 6" x 3" 1/4"	1/8" x 1/8" x 1/8" Screen	6
5	O-1262	Bearing Block	2
6	O-2261	Safety Valve Assy	1
7	O-1263-1	Cover Spring L.C.	1
8	O-1263-2	Cover Spring R.H.	1
9	O-1264	Hydro Liner	1
10		Splitter Pin	1

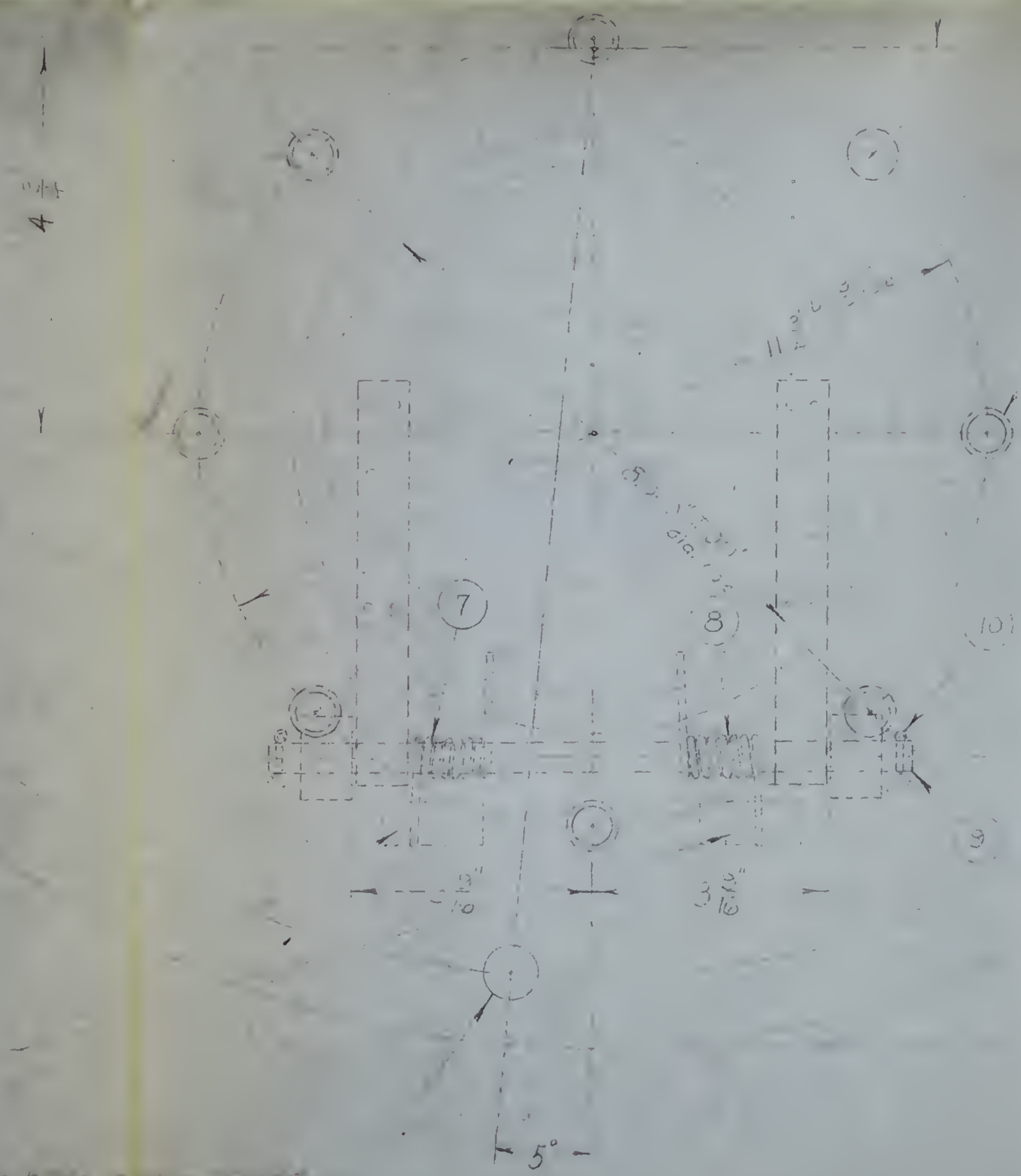
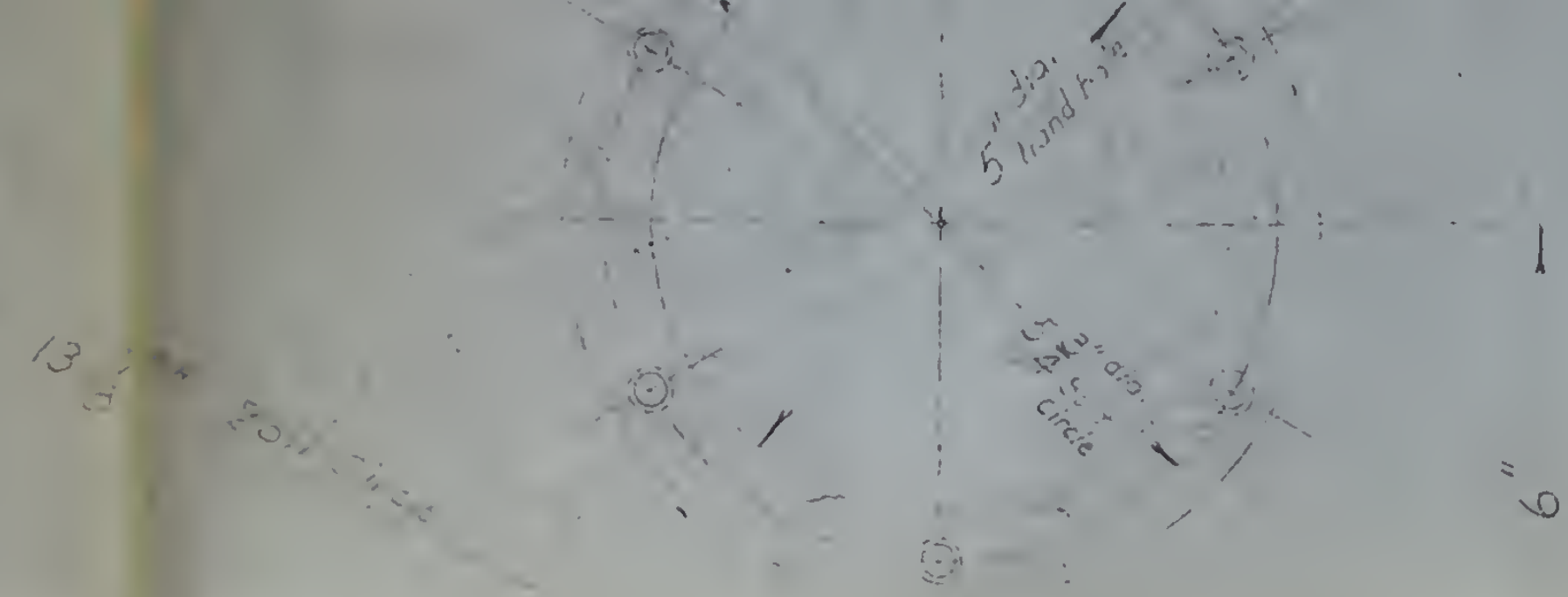
(3)

(4)

(2)

(1)

(6)



DRILLING - 3 - 1/2" V. 1  
3" dia.

Locate 1/4" hole in center of plate -  
To insure proper fit of plate  
with motor base plate

20 13/16" dia. holes in plate -



(1)

(6)

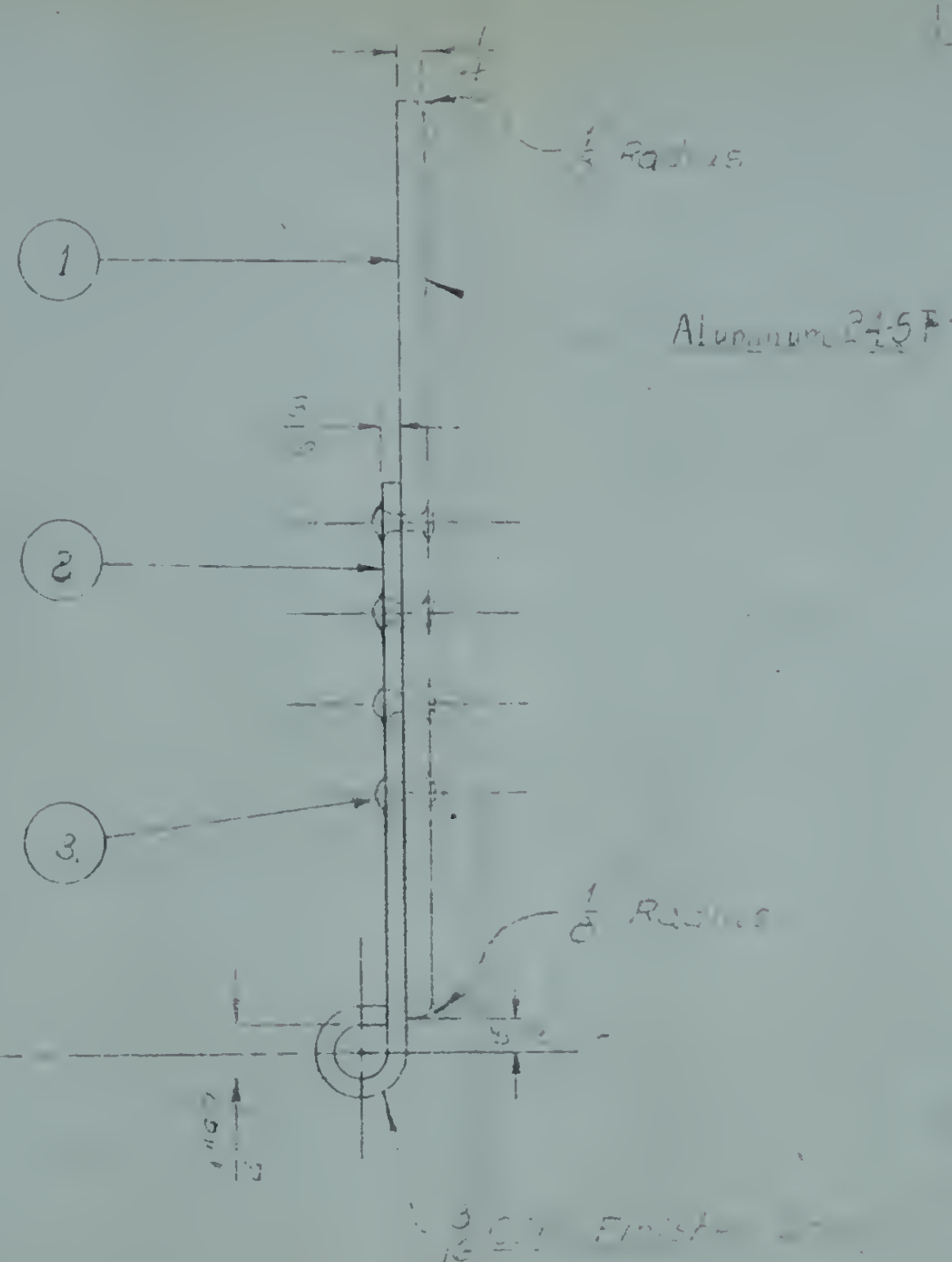
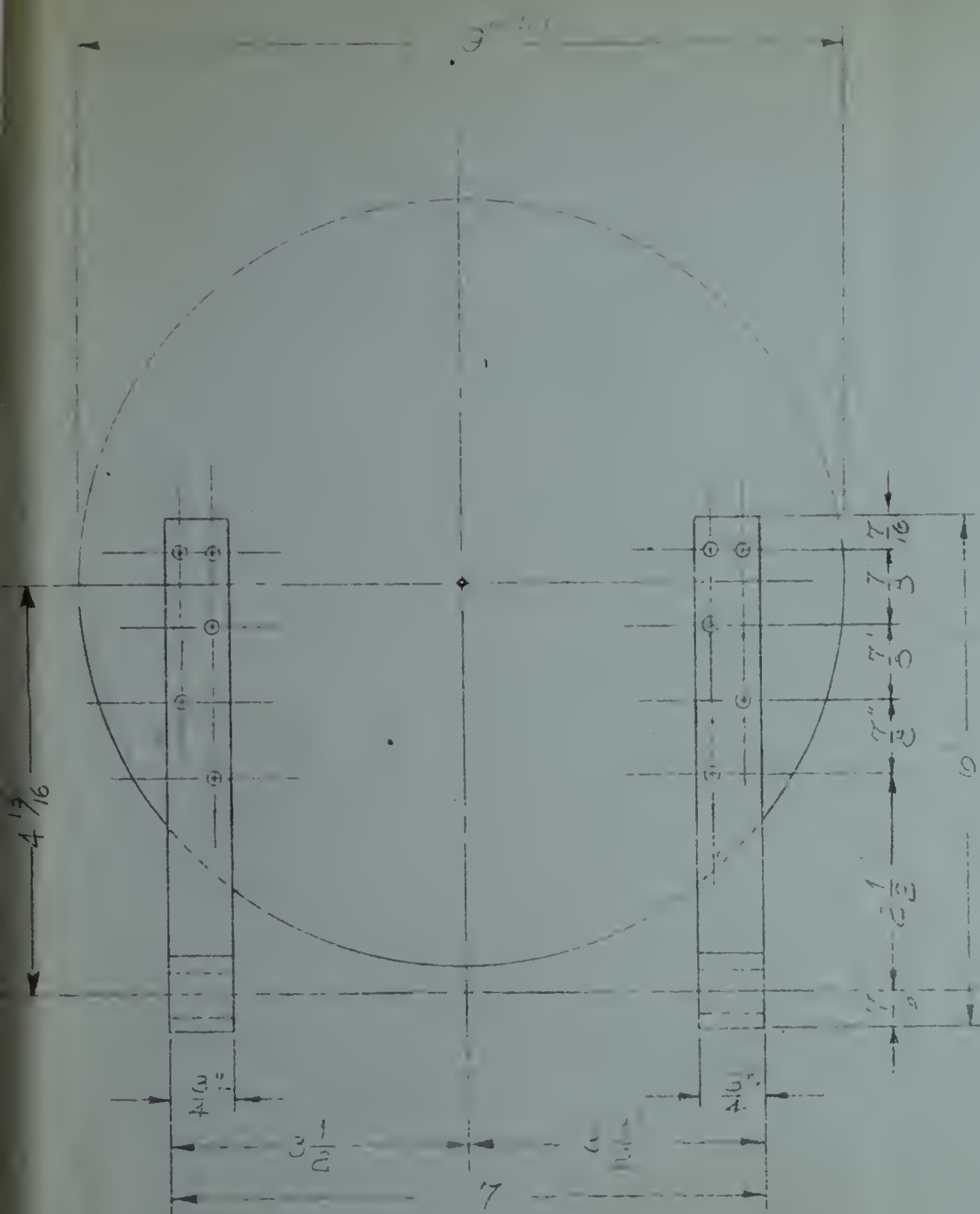


ASSEMBLY IN 5-11-49

NO.	MATERIAL	NO REQ
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GAS TUNNEL LABORATORY		
LOWER TUNNEL ADAPTER		
FLANGE ASSEMBLY		
DRAWN BY	BURMAN	DATE 5/11/49
CHECKED BY	J. W. R. L.	SCALE Half Size
NEXT ASSN	O-1266	4260

Fig. 4





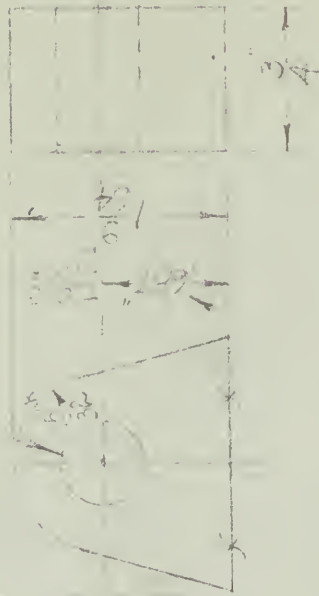
NO.	MATERIAL	NO. REQ.
1	C 2261	ENERGY COVER
2		RANGE
3	3 3/4 x 1/2	SUTTON 40 RYEN

NO.	MATERIAL	NO. REQ.
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GAS TURBINE LABORATORY		
SAFFETY VALVE		
DRAWN BY	1/2 inch	DATE 5/1/47
CHECKED BY	1/2 inch	DATE 5/1/47
NEXT ASSEM.	D 4250	D - 2261

FIG.5

cut 0.001" dia. + .0025  
 in a circle at distance 3/4 from center

-15°



INCOMPLETE

CONFIDENTIAL 2000

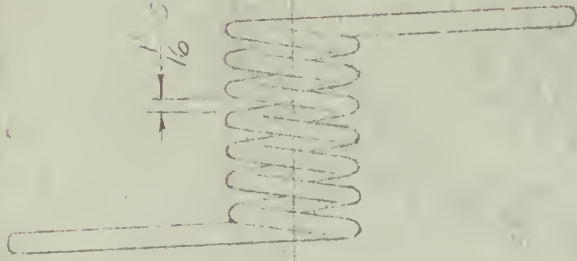
NO.	MATERIAL	NO. REQ
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GAS TURBINE LABORATORY		
BEARING BLOCK,		
SAFETY VALVE HANDLE		
DRAWN BY	DATE	5/10/49
CHECKED BY	SCALE	
NEXT ASSEM.	0-4200	06 1262

FIG. 6





6 1/2 full wire open winding



1/16 square wire open coils

110°

22"

1/16"

1 Same Spring as 13, Round 1

2 0.0375 J. Hard  
Spring steel wire Right Hand 1

NO.	MATERIAL	NO. REQ.
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GAS TURBINE LABORATORY		
SAFETY VALVE SPRINGS		
DRAWN BY	BURNS	DATE 5/10/49
CHECKED BY	Sm. H.	SCALE FULL
NEXT ASSEM.	O-4260	O-1263

is opposite hand (left hand wound)

Fig. 7



NO.	MATERIAL	NO. REQ.
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GAS TURBINE LABORATORY		
SAFETY VALVE GASKET Pk		
DRAWN BY	DATE	5/13/49
CHECKED BY	SCALE FULL	
NEXT ASSEM.		0-1264

Fig. 8





- Drill 6 holes,  $\frac{3}{16}$ " dia evenly spaced

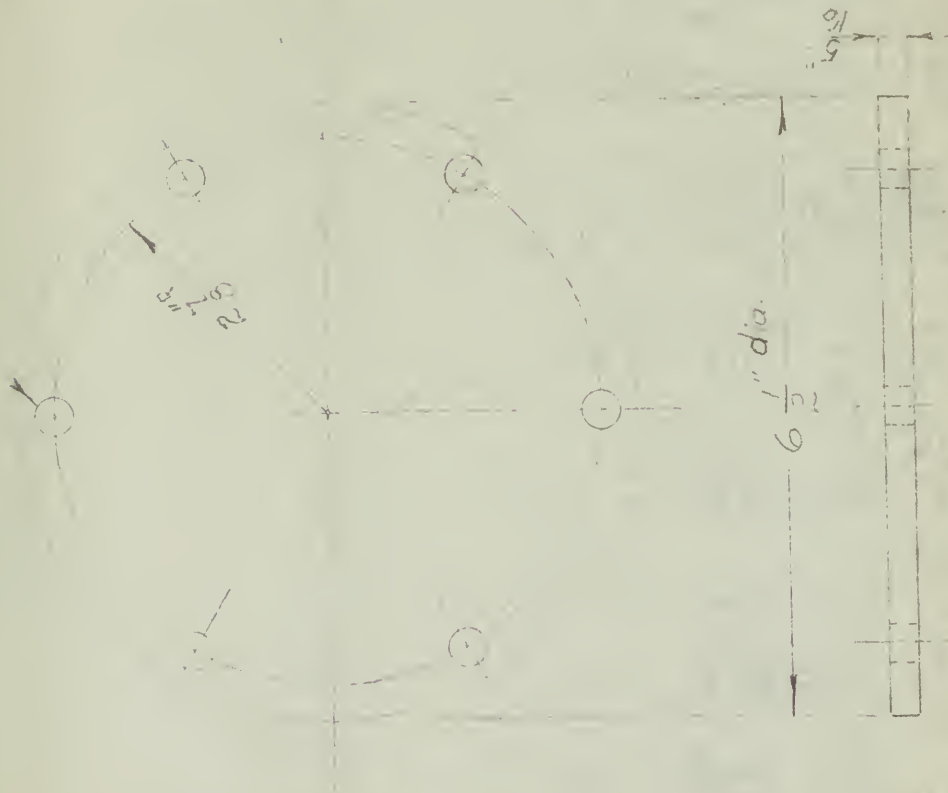


FIG. 9

5-10-300-0001 (REV)		1
NO.	MATERIAL	NO. REQ
MASSACHUSETTS INSTITUTE OF TECHNOLOGY GAS TURBINE LABORATORY		
HAND HOLE COVER PLATE, Lower Tunnel Adapter		
DRAWN BY P. U. M. S.	DATE 5/10/49	
CHECKED BY <i>W. S. H.</i>	SCALE HALF SIZE	
NEXT ASSEM. 0-4260		0-1265

$$\rightarrow \frac{1}{4} \rightarrow$$

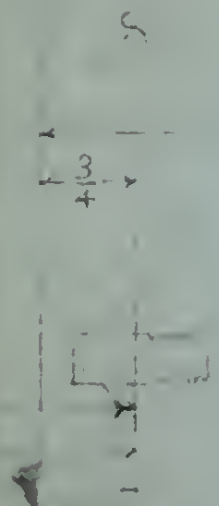
24' Welding Light W - 40' F 1000  
T<sub>2</sub>, no. 1000 e 11.12 V. 1000 on Equa



Neatly at E - Shows Part of  
90° East - True Part  
Part No. 3 - on Equator -

$$5\frac{1}{2} \pm \frac{1}{12}$$

12.175 + 100

$$11. \frac{1!}{1!} + \frac{1}{2}$$


DWL 000-2427  
E. 1. F. 1. R. 1. 1.

Down and Tidy  
back 6 Hole



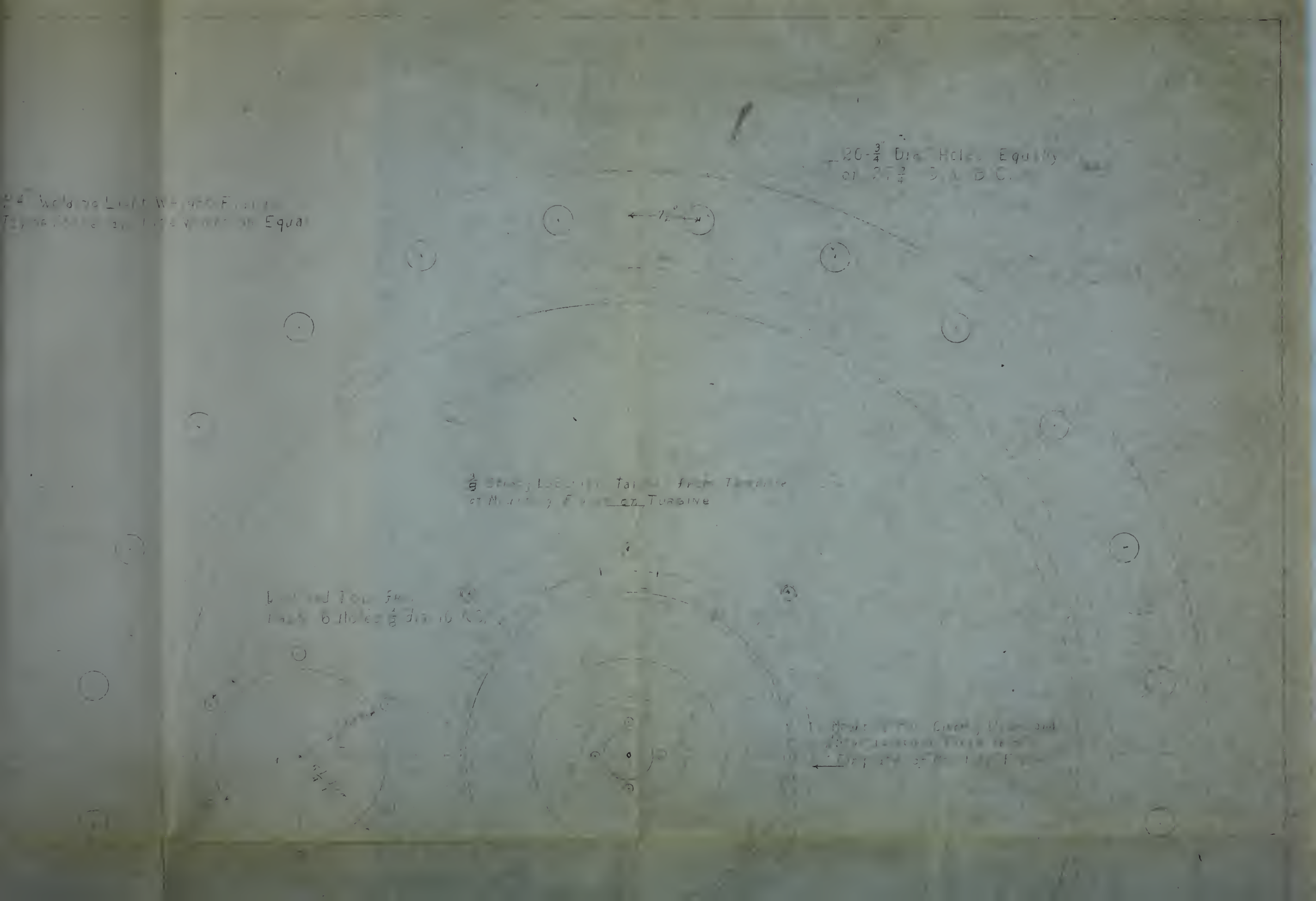
Weld to Left W-450 Flange  
 Type 304 SS, 1/2" thick, 18" Equidistant

20-3/4" Dia. Holes Equidistant  
 at 27 1/2" S.A.D.C.

3/8" Dia. 1/2" Long Tail from Turbine  
 at Mounting Flange of Turbine

6 Holes 1/2" Dia. 1/2" Long  
 1/2" Dia. 1/2" Long

Mounting Flange, 1/2" Thick and  
 1/2" Dia. 1/2" Long  
 1/2" Dia. 1/2" Long



151

8

6 8  
S. 100 2 41

100 2 41  
100 2 41

100 2 41  
100 2 41

100 2 41



1/4" hole

Cold Finished Steel

- 1 Turbine Mounting Flange 1
- 2 1/2" 24' Welding Collar 2

NO.	MATERIAL	NO. REQ.
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GAS TURBINE LABORATORY		
TUNNEL PIPING ASSEMBLY		
DRAWN BY	J. S.	DATE 5/2/45
CHECKED BY	W. S.	SCALE 1/2" = 1"
NEXT ASSEMBLY	D-4260 D-5259	OF 4266

Fig. 10





FIG. 11





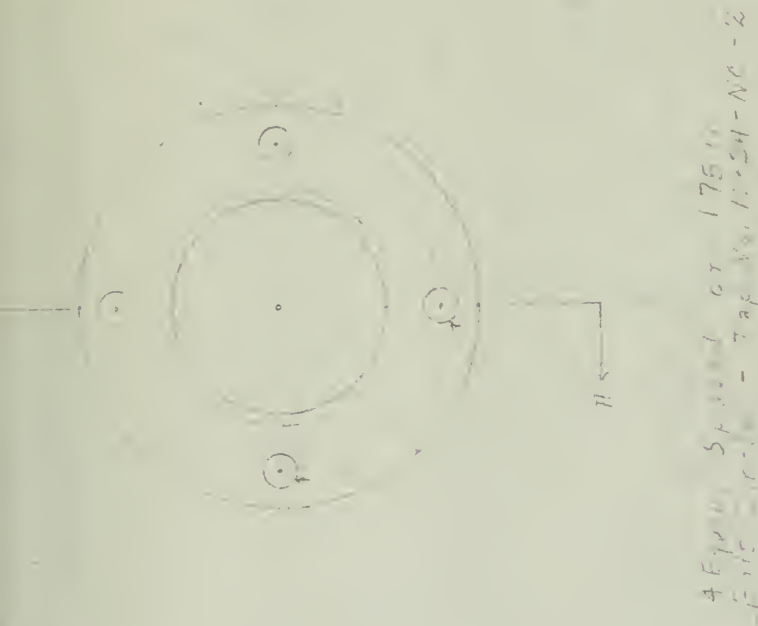
1" 1/16" - 1" R31

4-5/16" DIA. HOLE  
ON 1.75" DIA.

Fig. 12

NO.	MATERIAL	NO REQ
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GAS TURBINE LABORATORY		
FIFTH STATION		
9-16-57		
DRAWN BY	DATE	SCALE
CHECKED BY	FILE	
NEXT ASSEM.	O-2720	O-1268





7-C 1.75 inch diam.  
1.75 inch diam.  
1.06 inch diam.

Section M-M

1

NO.	MATERIAL	NO. REQ.
	ALUMINUM BRONZE	1
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GAS TURBINE LABORATORY		
FORWARD SHAFT EXTENSION		
SEAL		
DRAWN BY	DATE	5/10/47
CHECKED BY	SCALE	FULL
NEXT ASSEM.	0-5259	0 1269

Fig. 13





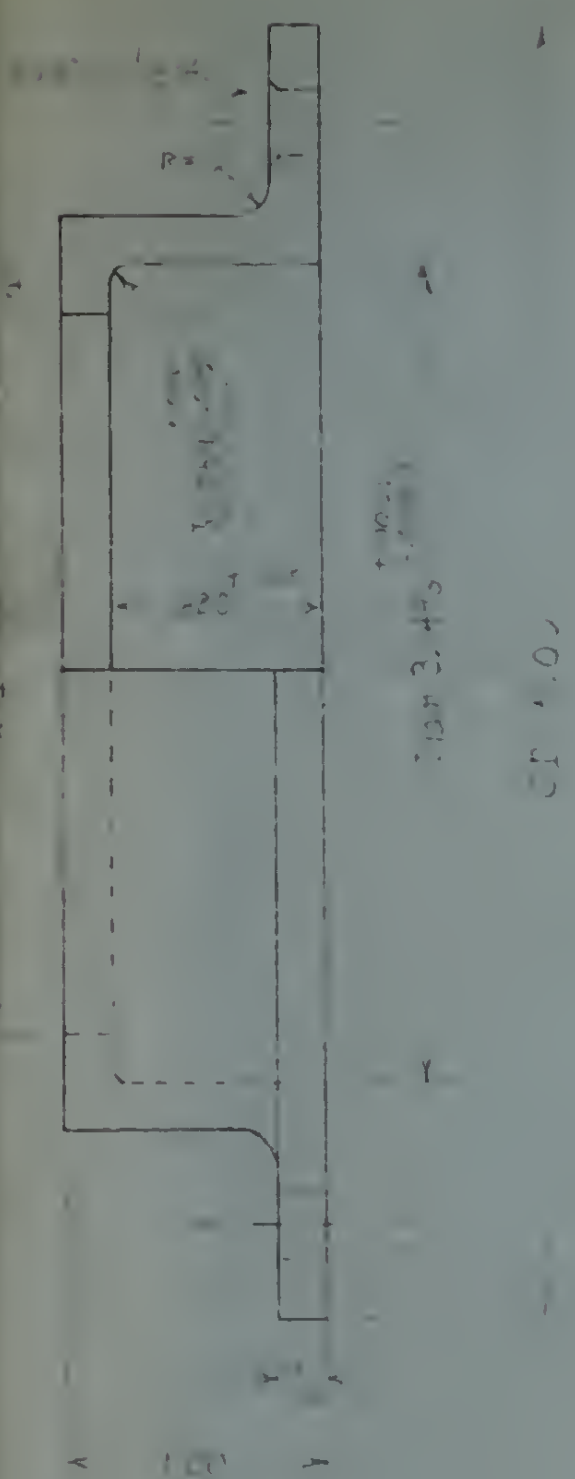
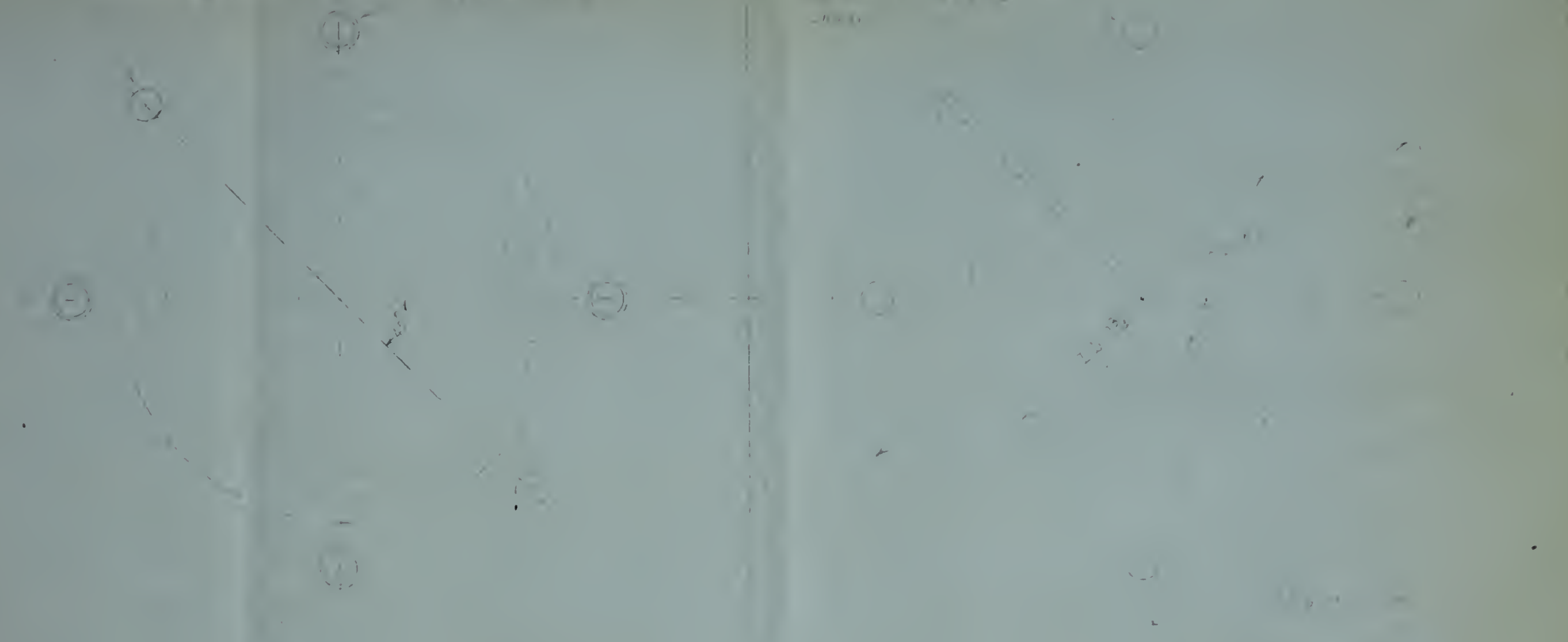


FIGURE 14 - SUBMIT (1/2\"/>

Drawn to scale 1/2\"/>

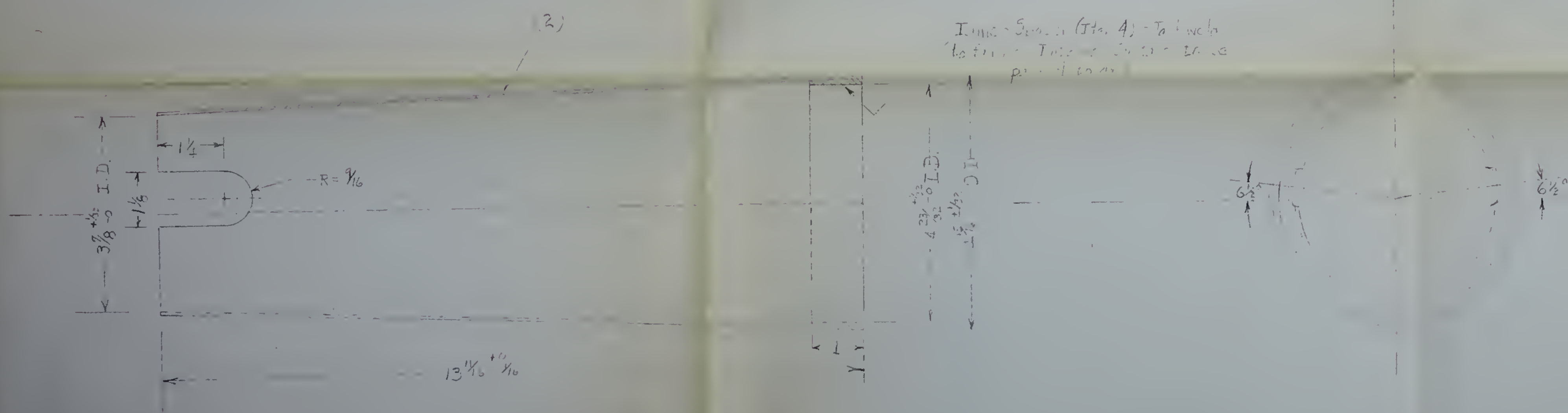
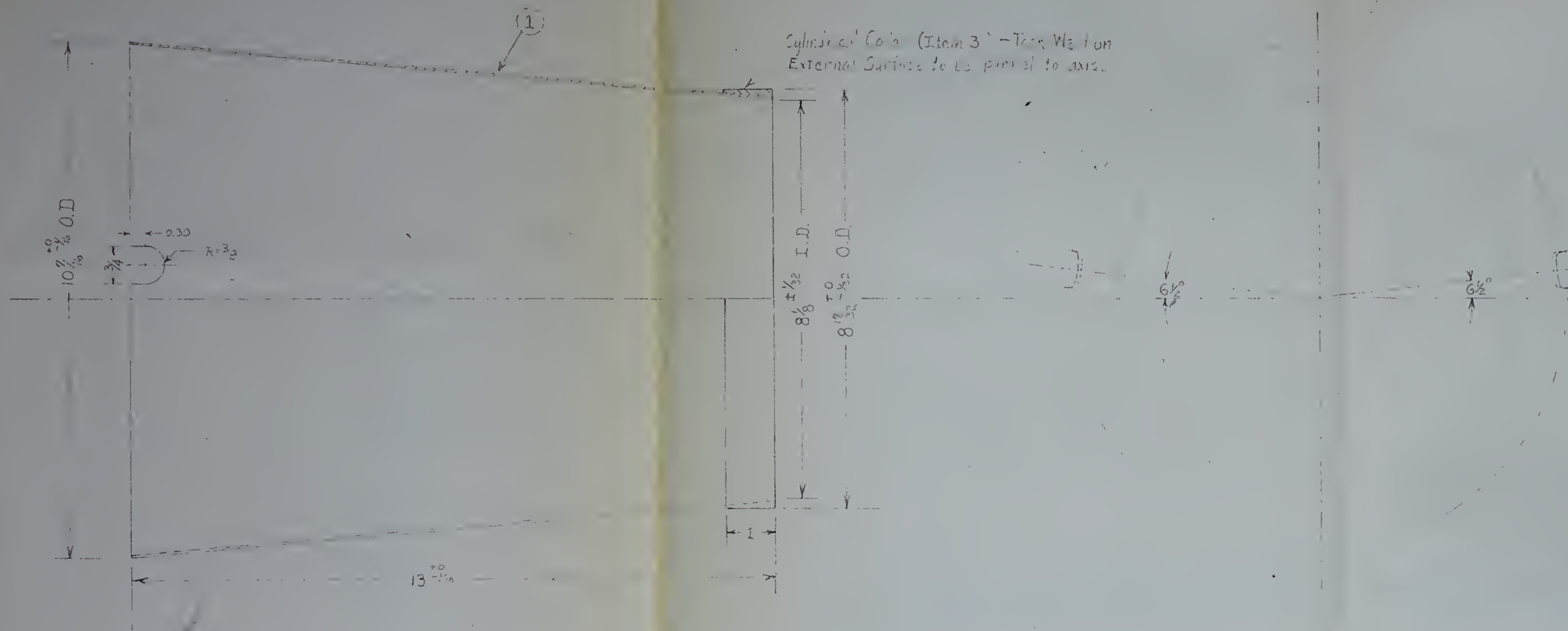


1	1/2\"/>
NO.	NATIONAL
MASSACHUSETTS DEPARTMENT OF TECHNOLOGY	
GAS TUBE LABORATORY	
Faint Flaming Hot	
in Silver	
DRAWN BY	DATE
CHECKED BY	SCALE
2270	

FIG. 14







4	Mild Steel	1
3	Mild Steel	1
2	Sheet Metal - 19 gauge	1
1	Sheet Metal - 19 gauge	1

NO.	MATERIAL	NO. REQ.
-----	----------	----------

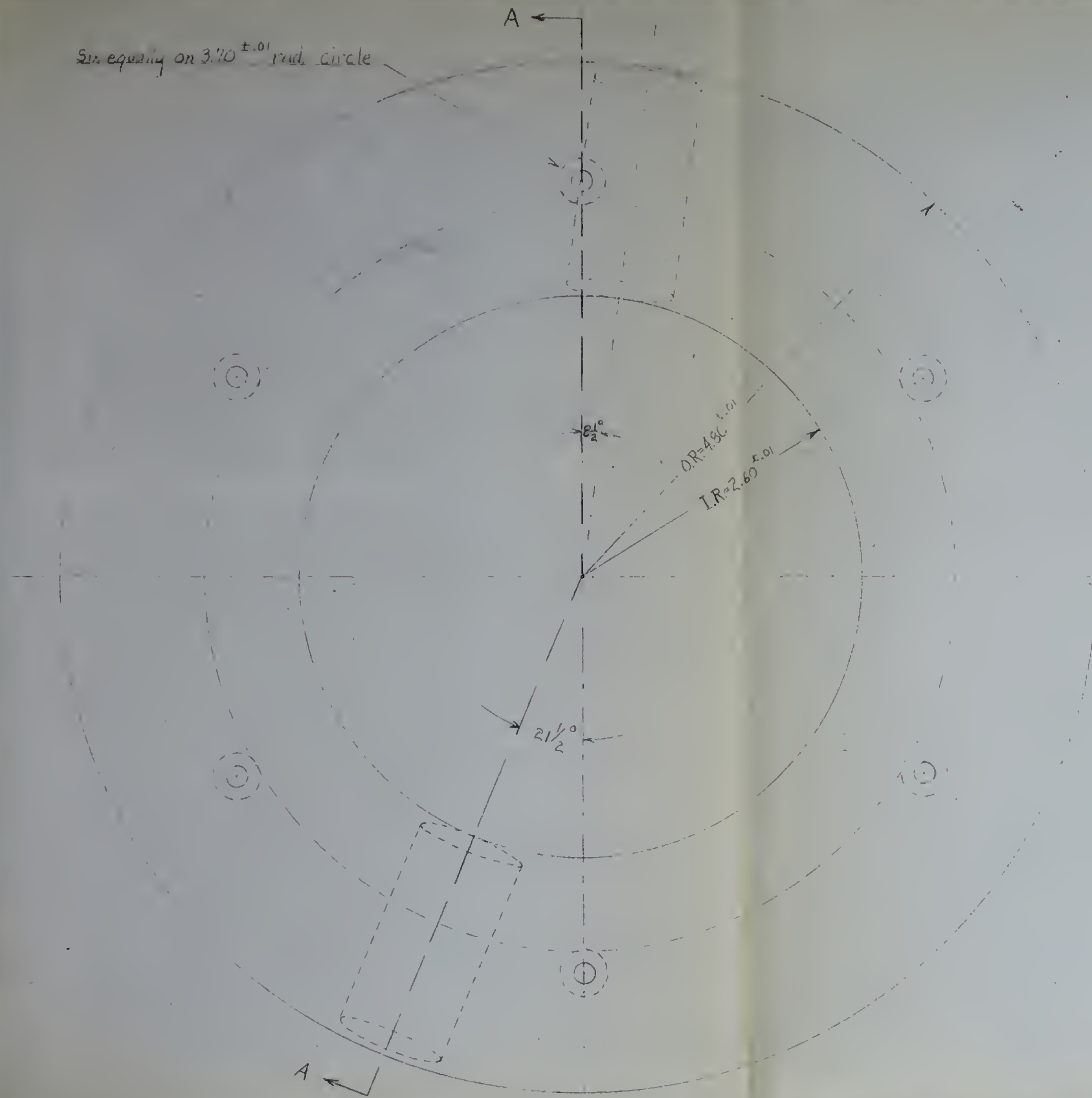
MASSACHUSETTS INSTITUTE OF TECHNOLOGY  
CHEMISTRY LABORATORY

INNER FAIRINGS

DRAWN BY: M. J. L. DATE: 5/10/49  
CHECKED BY: SCALE: 1/2" = 1"  
ASSEMBLY: 0-5209 0-2271

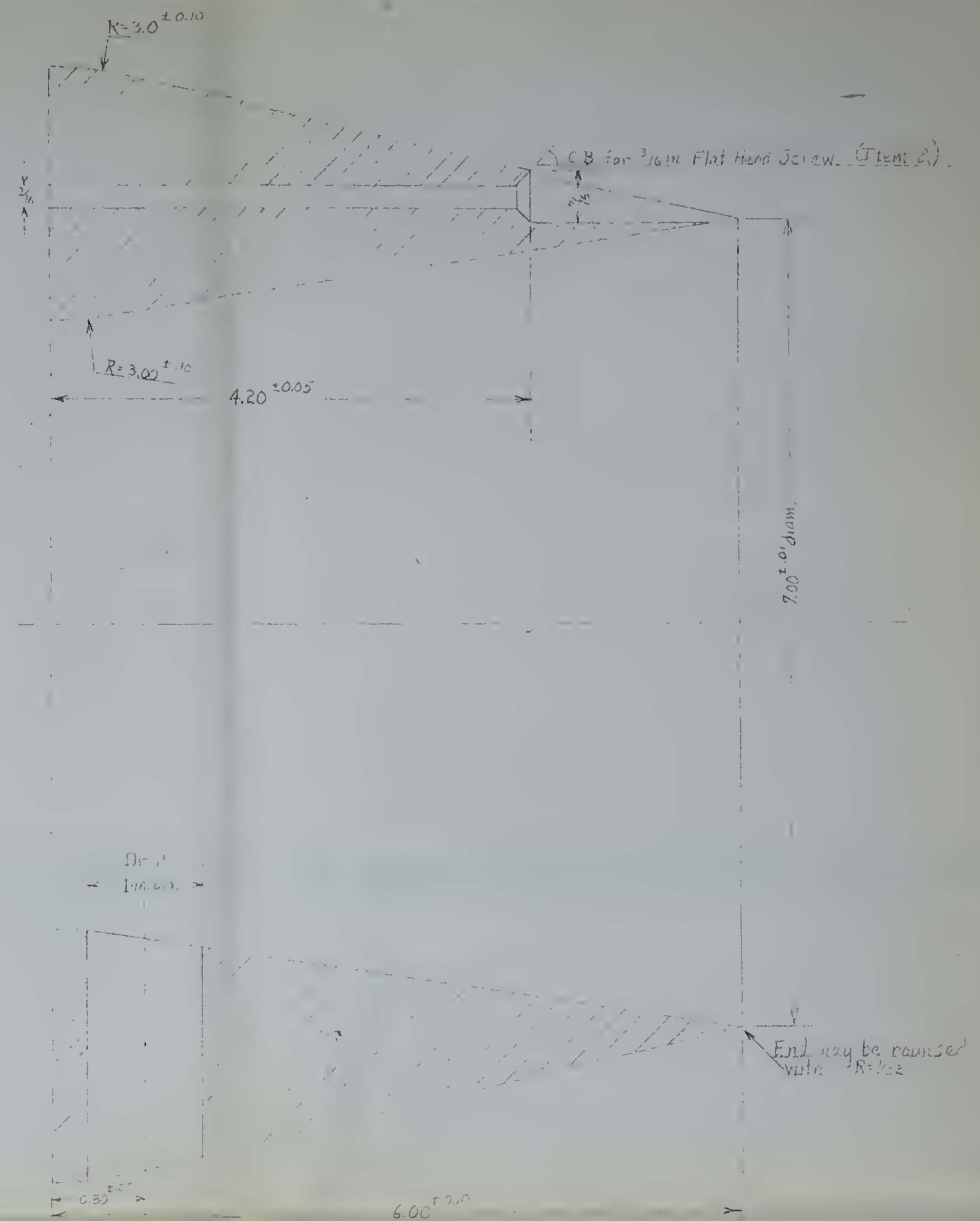
Fig 15





Front End View  
(Item 1)

Mat'l: Wood or Any Metal, as desired.



Section A-A

2	4 1/2 x 3/8 Flat Hd Screw (NF Tha.)	6
1	Wood or Any Metal	1
NO	MATERIAL	NO. REQ.
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GAS TURBINE LABORATORY		
TAIL CONE		
DRAWN BY	Howson	DATE 5/10/47
CHECKED BY		SCALE Full Size
NEW YORK	Part 16 of 21	0-2272
	Ref. 10-5-1	

Fig. 16

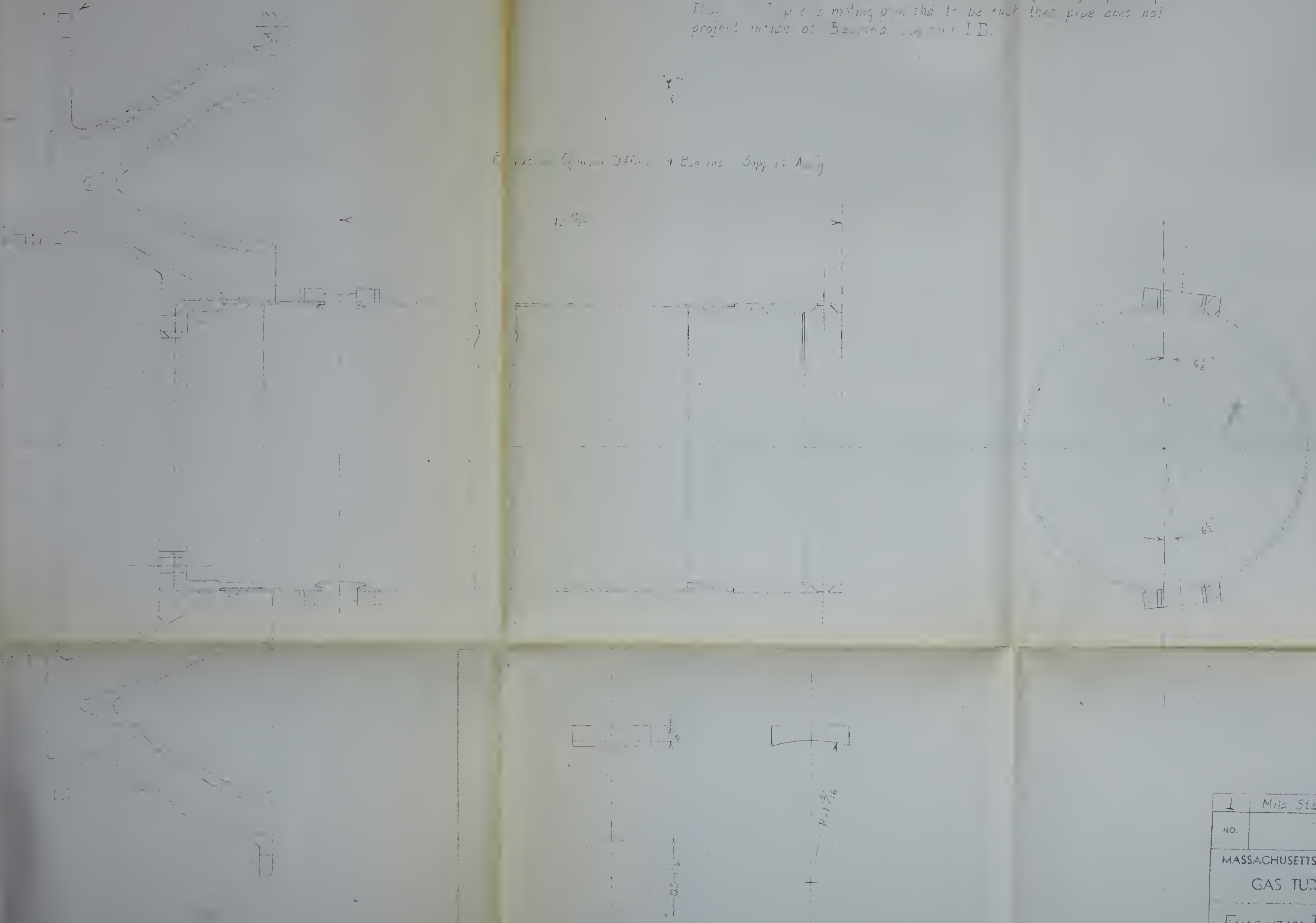




Top of Duct is on 1/2" ...  
inside 3/4" ...

Assembly Note: Exhaust Pipe Adapters to be installed to Bearing Support  
in position shown prior to installation for 3/4" (nominal) Taper Pipe  
The ... of ... must be such that pipe does not  
project inside of Bearing ... ID.

Exhaust Pipe Adapter & Bearing Support Assembly

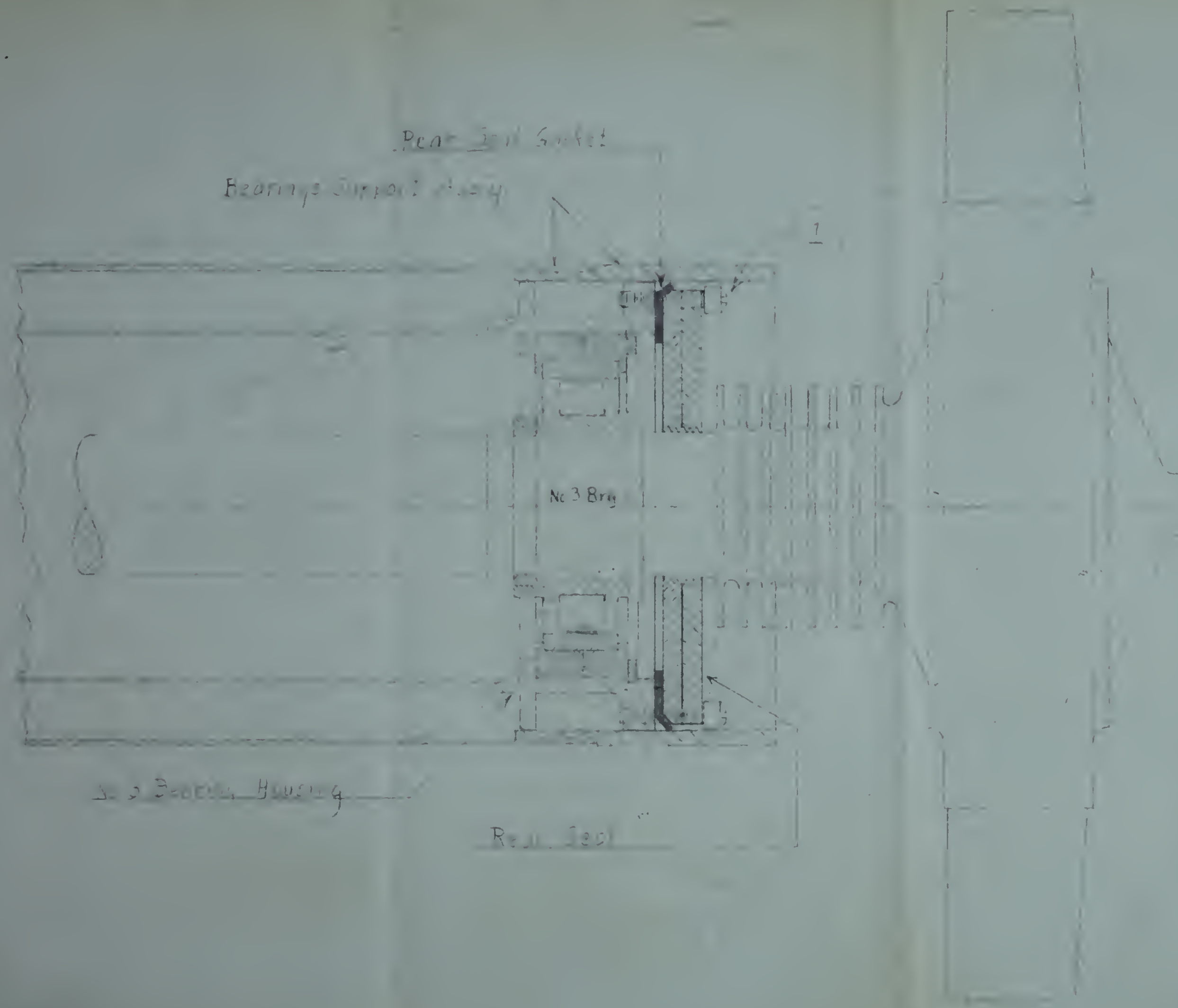


Exhaust Pipe Adapter - Blank, Details (Item 1)

1	Mild Steel	2
NO.	MATERIAL	NO. REQ.
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GAS TURBINE LABORATORY		
EVACUATION PIPE INSTALLATION and ADAPTER DETAILS		
DRAWN BY <u>M. J. ...</u>		DATE <u>5/10/49</u>
CHECKED BY <u>...</u>		SCALE <u>Full Size</u>
PART ASS. NO. <u>0-5259</u>		<u>0-3273</u>

Fig. 17





1	Fillister Hd Cap Screw, 1/4" 5006-24K	4
NO.	MATERIAL	NO REQ
MASSACHUSETTS INSTITUTE OF TECHNOLOGY GAS TURBINE LABORATORY		
REAR SEALS - SUB ASSEMBLY		
DESIGNED BY	W. J. ...	DATE 5/10/49
CHECKED BY	...	SCALE 1/4" = 1"
PRINT ASSEM.	C-2274	Q - 2274

FIG. 18



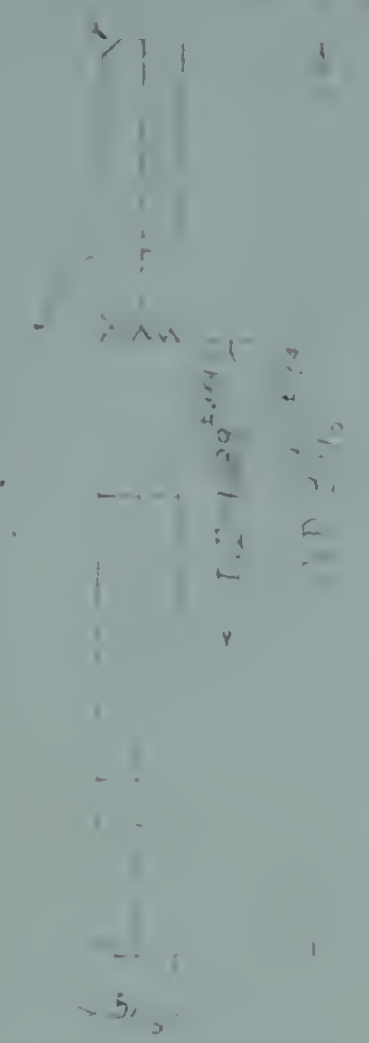
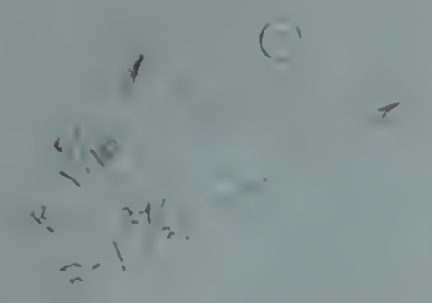


Note: Seal gasket must be replaced with new gasket  
 to avoid leaks. Seal gasket must be replaced with new gasket.

Date: 10/10/10  
 Page: 1 of 1

1-10-10

1-10-10



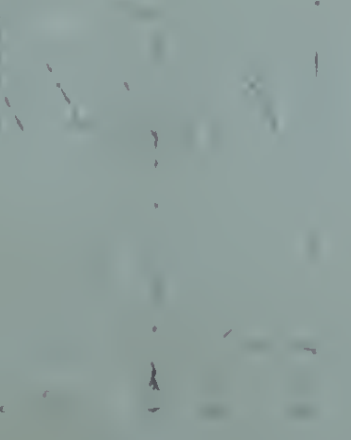
GASLET (10/10)

SCALE (10/10)

Note: Gasket must be replaced with new gasket  
 Circumferential Edge (C/E) must  
 be truly cylindrical in shape  
 after installation.

Note: Gasket must be replaced with new gasket  
 Circumferential Edge (C/E) must  
 be truly cylindrical in shape  
 after installation.

Note: Gasket must be replaced with new gasket  
 Circumferential Edge (C/E) must  
 be truly cylindrical in shape  
 after installation.



MASSACHUSETTS DEPARTMENT OF REVENUE	
GAS ENGINE LABORATORY	
RECEIVED 10/10/10	
DATE	10/10/10
TIME	10:10
BY	10/10/10
FOR	10/10/10
TO	10/10/10
FROM	10/10/10
REMARKS	10/10/10
10/10/10	

Fig. 19









1.000  
 1.000  
 1.000

1.000  
 1.000  
 1.000

1.000  
 1.000  
 1.000

1.000  
 1.000  
 1.000

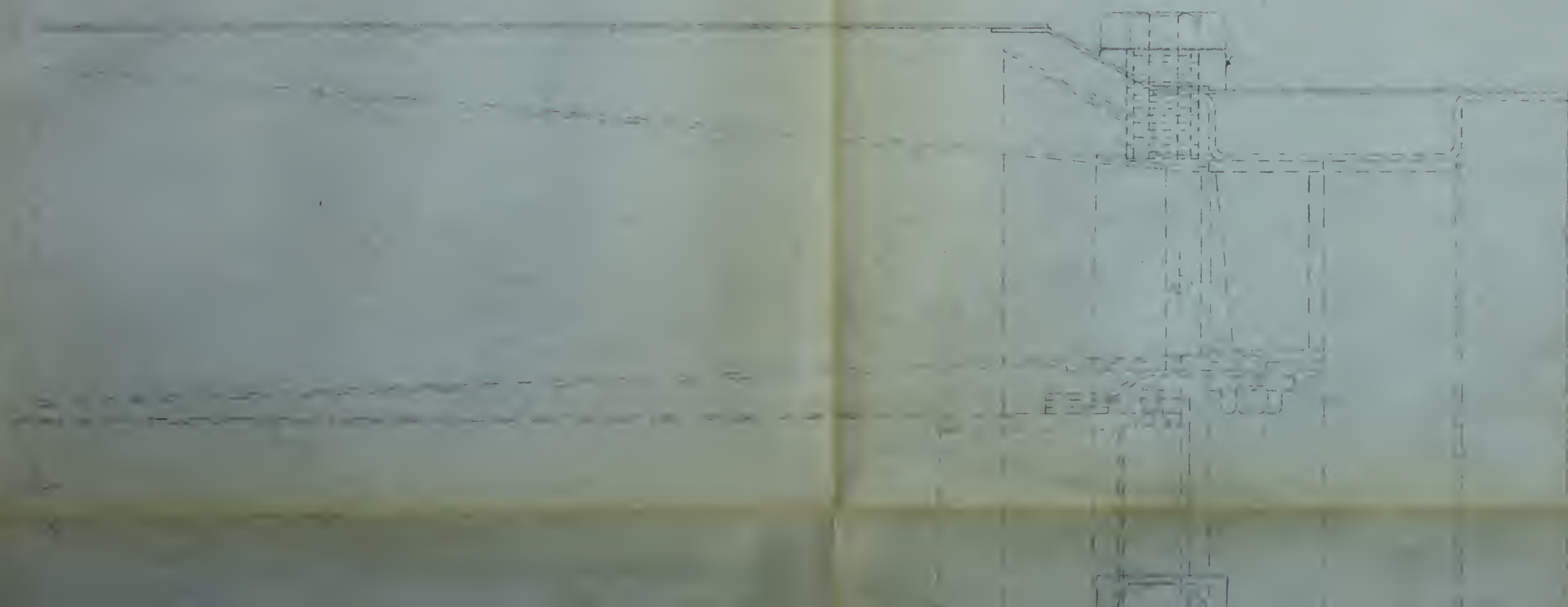
3.250  
 5.830 ± .003  
 5.6275 ± .004  
 6.125 ± .002

Machine and tool  
 for the purpose of

Fig. 21

NO.	MATERIAL Aluminum 2024-T3	NO. REQ.
MASSACHUSETTS INSTITUTE OF TECHNOLOGY GAS TURBINE LABORATORY		
NOSE CONE		
DRAWN BY [Signature]		DATE 5/10/42
CHECKED BY [Signature]		SCALE Full
NEXT ASSEM. 0-5259		0 2277

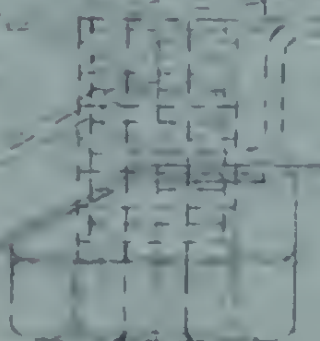
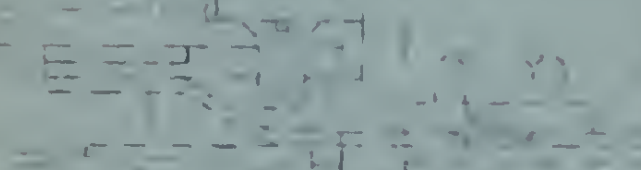
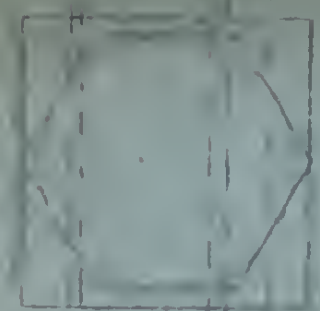
Sketch of a building with a chimney  
and a small structure in front  
of it.



Building Lab







SECTION - 1

1/2" = 1'



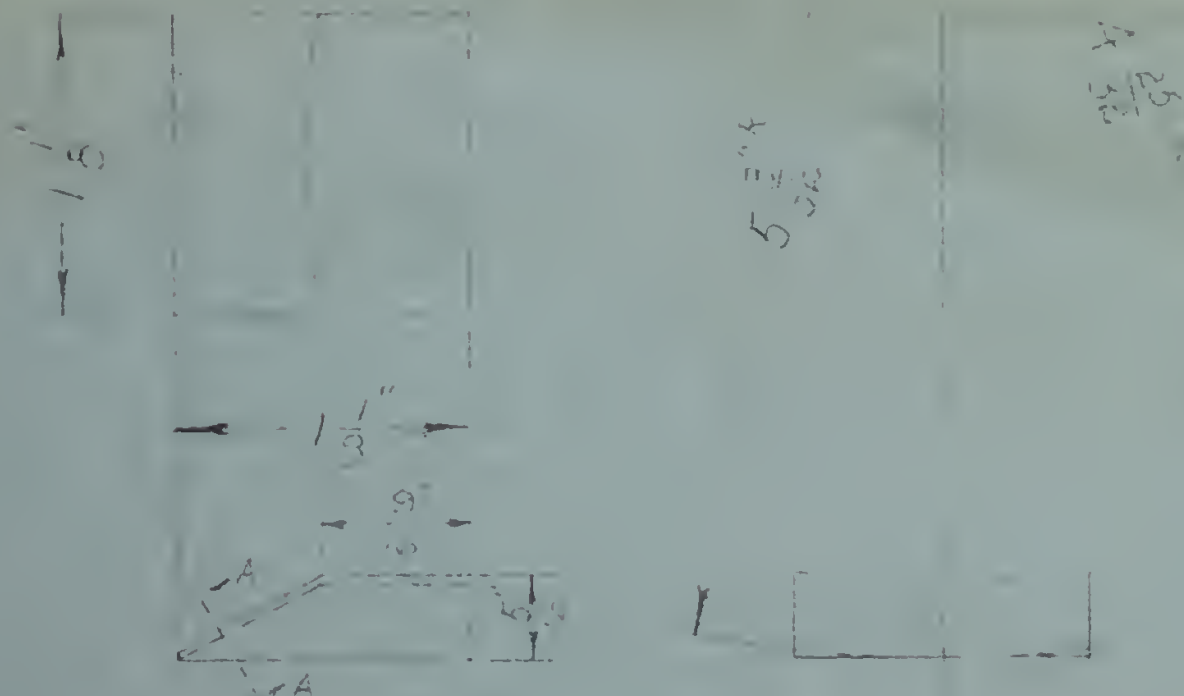
REVERSE SIDE



1	DATE	Q-4275
2	DATE	Q-4275
NO.	MATERIAL	(MATERIAL)
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GMA MECHANICAL LABORATORY		
INSTRUMENTATION DIVISION		
ELECTRONIC ENGINEERING		
TESTED BY	DATE	5/12/58
APPROVED BY	DATE	5-12-58
		Q-4275

FIG. 22





SECTION A-A

SECTION A-A

FIG. 24

1	Mild Steel	3
NO.	MATERIAL	NO. REQ.
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GAS TURBINE LABORATORY		
INSTRUMENT ACCESS PAD, BLANK.		
COMBUSTION CHAMBER		
DRAWN BY	Burns	DATE 5/11/49
CHECKED BY	Morse	SCALE Full Size
NEXT ASSEM.	D-4278	D- 1280





25  
30

25  
30



E

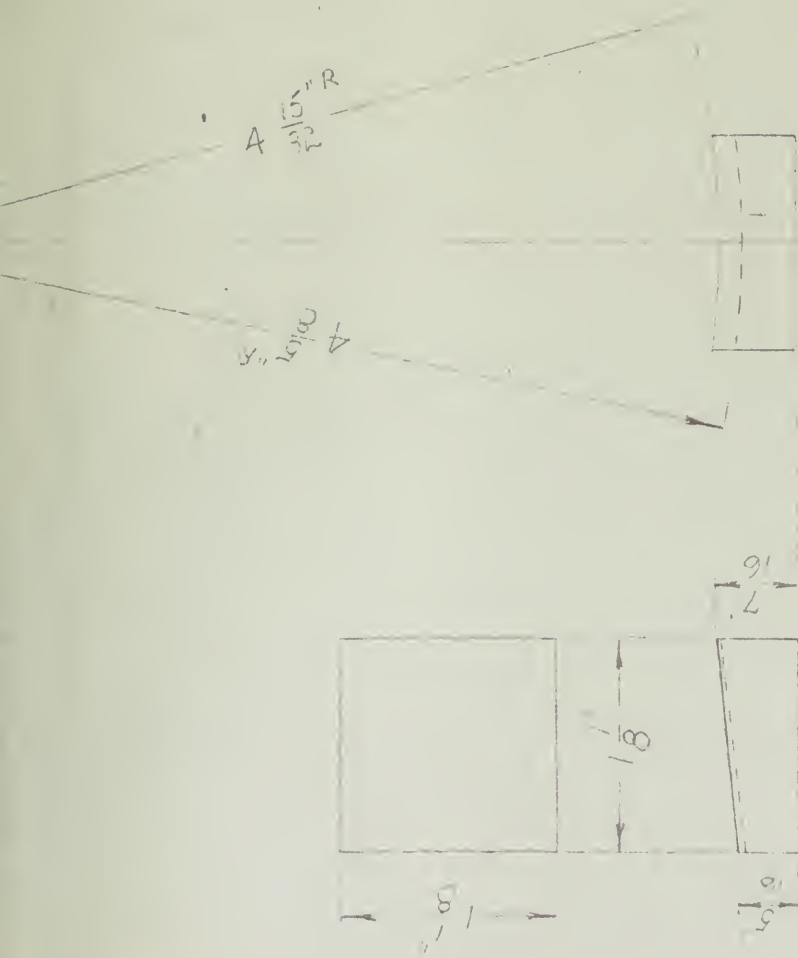
SECTION A-A

SECTION B-B

1	Mild Steel	3
NO.	MATERIAL	NO. REQ
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GAS TURBINE LABORATORY		
INSTRUMENT ACCESS PAB, BLANK,		
COMBUSTION CHAMBER		
DRAWN BY	Burns	DATE 5/11/49
CHECKED BY	Morgan	SCALE Full Size
NEXT ASSEM.	0 4278	0- 1280

Fig. 24



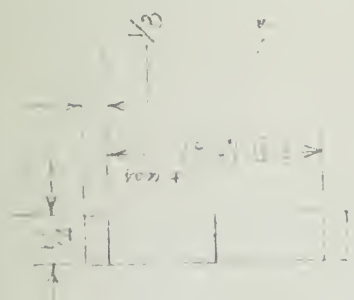


NO.	MATERIAL Cold Finished Steel	SHEET 4
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GAS TURBINE LABORATORY		
INSTITUTE OF TECHNOLOGY		
BLANK, TAIL PIPE		
DRAWN BY BURN	DATE 5/10/49	SCALE FULL
CHECKED BY		
NEXT ASSEMBLY 0-3282 0-12		

Fig. 25





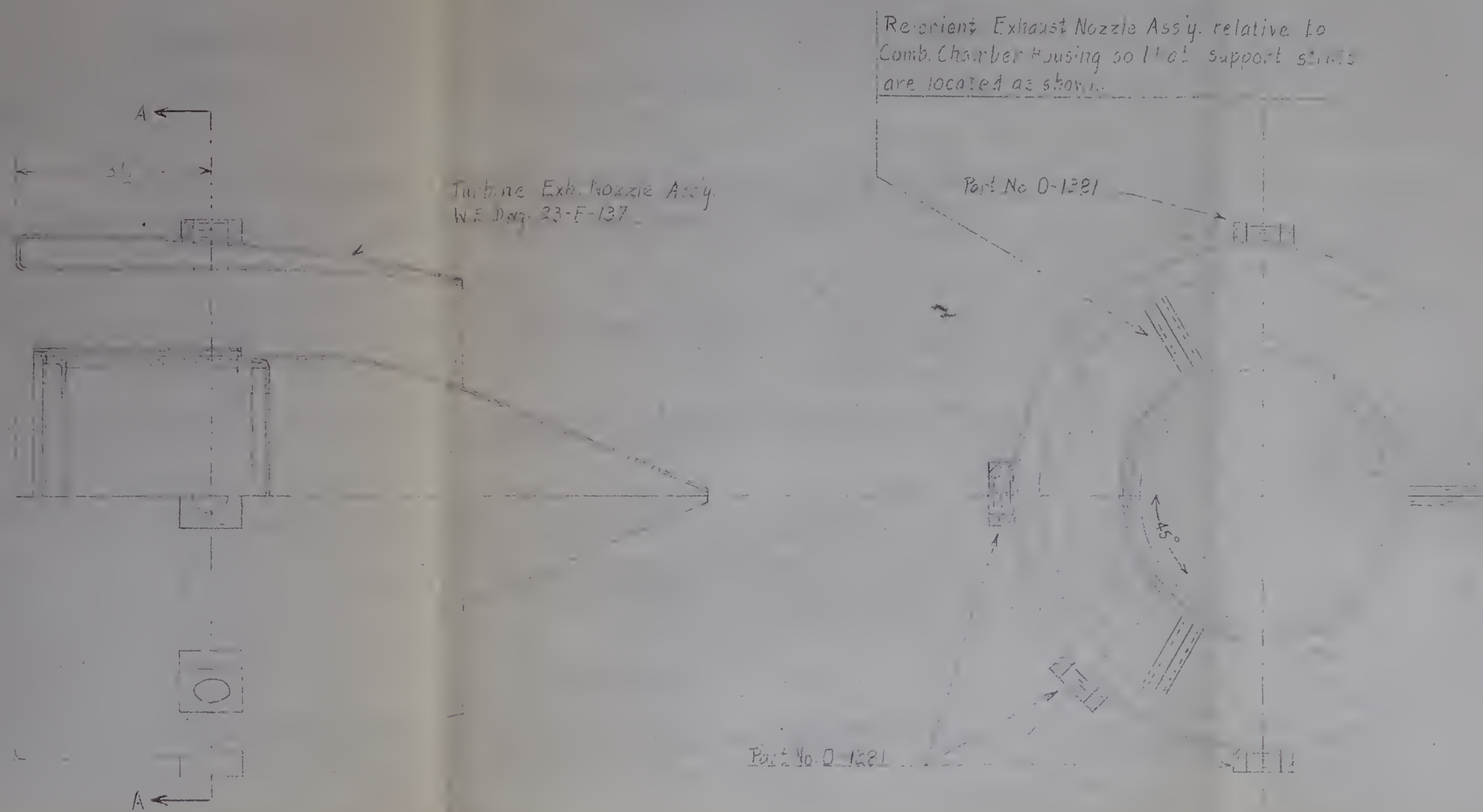


To be shown on Turbine Case  
 Sectional View, P1-03-000  
 after assembly - Turbine Case

1	M.S. Steel	1
NO.	MATERIAL	NO. REQ.
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GAS TURBINE LABORATORY		
BEARING RETAINER RING		
DRAWN BY	DATE	7/1/54
CHECKED BY	SCALE	1/1/54
NEXT ASSEMBLY	Q-2000-5	Q-1000-5

Fig. 26





SECTION A-A

Showing Location for Instrument Access  
 Port (Part No. O-1281) on Turbine Exhaust  
 Nozzle (W.E. Dwg. 23-F-137)  
 Silver-Solder or Braze Pac Blocks into place  
 to provide location for Inst. Probe Bushings  
 (Part No. O-1279) Drill through all members  
 to an stream. After final assembly trim  
 Bushing end to coincide with outer boundary  
 of air stream

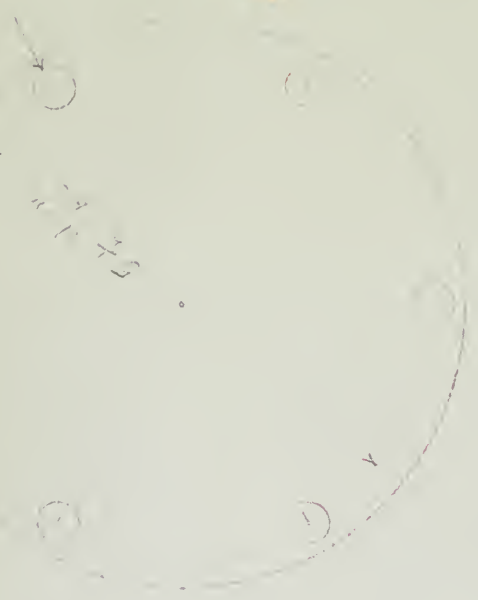
NO.	MATERIAL	REQ.
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GAS TURBINE LABORATORY		
INSTRUMENT ACCESS LOCATION		
TAIL - PIPE		
DRAWN BY	1/11/52	DATE 5/1/52
CHECKED BY		SCALE 1/11/52
NEXT ASSEM. C-5259		O-3283

Fig. 27





12 HOLES  $\frac{3}{32}$  DIA. EVERY 3 PITCH



12 DIA.

(1)



(2)

FIG. 28

1. <u>Mild Steel</u>	1	
2. <u><math>\frac{1}{16}</math> IN. MIPRAGE</u>	1	
NO.	MATERIAL	NO REQ
MASSACHUSETTS INSTITUTE OF TECHNOLOGY		
GAS TURBINE LABORATORY		
HAND. NO. 1 COVER AND GASKET		
TUBING: 1/2 IN. DIA. 1/4 IN. WALL		
DRAWN BY <u>2</u>	DATE <u>5/12/49</u>	
CHECKED BY <u>W. H. H. H.</u>	SCALE <u>1/2 IN. = 1 IN.</u>	
NEXT ASSEM. <u>0-4266</u>		<u>1286</u>













29 MAY 69  
8 JUN 71

17520  
19264

13666

Thesis  
M52

Mercer

Design of a test installation for investigation of Reynolds' number effect on gas turbine performance.

29 MAY 69  
8 JUN 71

17520  
19264

Thesis  
M52

13666

Thesis  
M52

Mercer

Design of a test installation for investigation of Reynolds' number effect on gas turbine performance.

Library  
U. S. Naval Postgraduate School  
Monterey, California



thesM52

Reynolds' number effect on gas turbine p



3 2768 001 88610 4

DUDLEY KNOX LIBRARY